THE DEEP WATER GAS CHARGED ACCUMULATOR AND ITS POSSIBLE REPLACEMENTS

A Thesis

by

MEHDI MIR RAJABI

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

December 2004

Major Subject: Petroleum Engineering

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Approved as to style and content by:

Hans C. Juvkam-Wold (Chair of Committee)	Jerome J. Schubert (Member)
Charles P. Aubeny (Member)	Stephen A. Holditch (Head of Department)

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ABSTRACT

The Deep Water Gas Charged Accumulator and Its Possible Replacements. (December 2004)

Mehdi Mir Rajabi, B.Sc., Sahand University of Technology (Iran)

Chair of Advisory Committee: Dr. Hans C. Juvkam-Wold

Blowout preventers are designed to shut in a well under pressure so that formation fluids that have moved into the wellbore can be contained and circulated out while continuous control of the well is maintained. Control Systems for the BOPs are of necessity highly efficient hydraulic systems. The objective is to operate functions, such as closing rams, on the BOP stack in as short a time as possible. Supplying enough volume of pressured hydraulic fluid to operate those emergency functions is essential. To have the necessary quantity of control fluid under pressure requires storing this fluid in accumulators. These accumulators operate by the expansion and compression of nitrogen gas that is separated from hydraulic fluid by either rubber bladders or pistons.

Accumulators are used both on the surface and at the seafloor. As long as you use accumulators on the surface or in relatively shallow waters, you may not have a problem with the volume of hydraulic fluid capacity of gas charged accumulators. The problem may arise when the wellhead is at water depth of more than 3500 ft. In deep water drilling, the accumulators should be placed on the subsea blowout preventer stack to reduce hydraulic response times and provide a hydraulic power supply in case of interruption of surface communication. Accumulators are also used in subsea production control systems to provide local storage that allows smaller line sizes in control umbilicals. Hydraulic fluid capacity of an accumulator drops to 15% of its capacity on the surface and even less, depending on the water depth. A large number of accumulators are needed to perform BOP functions that could have been done by just a few of them on the surface or at relatively shallow water depth.

Gas inside gas charged accumulators does not behave like an ideal gas as we go to very deep water, due to high hydrostatic pressure at that water depth. The higher the ambient pressure, the more the gas behaves like a real gas rather than an ideal gas and the lower the fluid capacity of the accumulators. Compressed gas has energy in it, and can release this energy at the time desired, that’s why it is used in accumulators. Now, we have to look for something that is able to store energy, but unlike the nitrogen, its functionality must not be affected by the increasing hydrostatic pressure of water as a function of water depth. Springs and heavy weights will be discussed as two options to replace nitrogen in accumulators. Efficient deep water accumulators would reduce the number of accumulators required in deepwater and cut the cost of the project. With the advent of such efficient accumulators, we can hope that one of the numerous problems of deepwater drilling has been solved and we can think of drilling in even deeper waters.

To My Parents

ACKNOWLEDGMENTS

I would like to take this opportunity to express my sincere appreciation to the people who have assisted me throughout my studies.

I would specifically like to thank my advisor, Dr. Hans C. Juvkam-Wold, for his guidance and encouragement throughout my research. His patience over the period of doing my research is also greatly appreciated.

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CHAPTER I

INTRODUCTION

Since the 1960's gas charged accumulators have been placed on subsea blowout preventers to reduce hydraulic response times and provide a local hydraulic power supply in case of interruption of surface communication. Accumulators are also used in subsea production control systems to provide local storage that allows smaller line sizes in control umbilicals.[1] Usable Fluid, which is declared as the amount of pressurized liquid that an accumulator can hold, noticeably decreases as drilling and subsea production moves to ever-deeper waters so that a large number of accumulator bottles is needed to store liquid required to do close and open functions in that depth of water. This issue of gas charged accumulators introduces itself as one of numerous obstacles to ultra-deepwater drilling technology. This behavior of accumulators is in part because of non-ideal behavior of compressed gas, usually nitrogen, in high ambient pressure at the sea floor where accumulators are located. Even if, nitrogen behaves like an ideal gas, the volume of usable fluid decreases, since the hydraulic fluid exhausts to the sea-water to reduce the length of umbilicals and pressure drop. So, the calculation of usable fluid should compensate for the hydrostatic pressure of water depth where hydraulic fluid is supposed to exhaust. Assuming nitrogen as an ideal gas, the following formula is used for calculation of usable fluid at the water depth of D_w.

$$V_U = \left(\frac{P_n + 0.445D_w}{P_{\min} + 0.445D_w} - \frac{P_n + 0.445D_w}{P_{\max} + 0.445D_w} \right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1.1)$$

With the constant P_n, P_{min}, P_{max}, and V_{ac}, it is mathematically clear that the value of right hand side of the equation above decreases when the value of D_w, which is a positive value, increases. **Fig.1.1** shows how the volume of usable fluid decreases as water depth

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increases. This graph is plotted for a 15-gallon bladder accumulator (V_{ac} = 13.7 gal.) with a maximum working pressure of 5,000 psi, minimum working pressure of 2,000 psi, and a precharged pressure of 1,800 psi.

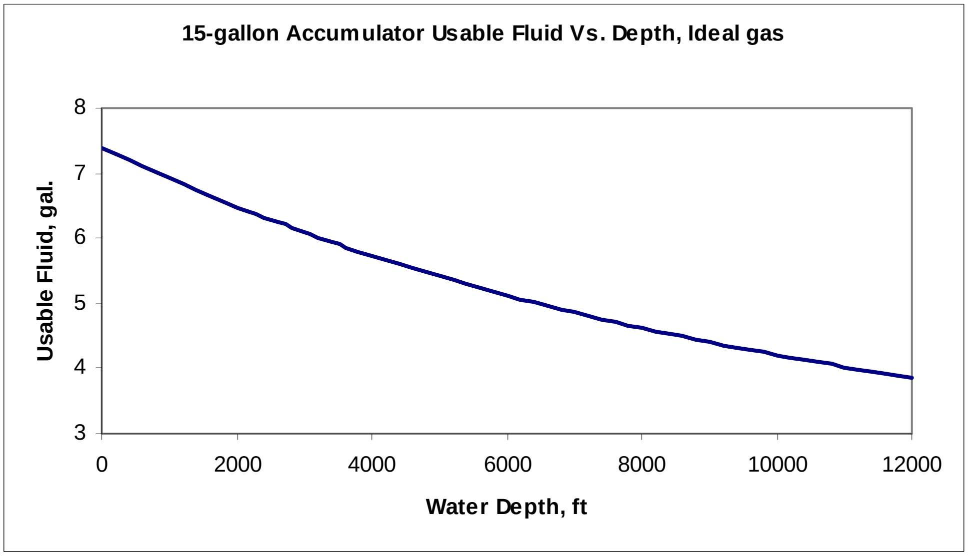


Fig. 1.1- Usable fluid decreases as water depth increases.

Calculations giving the usable fluid of accumulators are based on the gas equation of state. Nitrogen, a common gas used in accumulators, behaves like an ideal gas as long as ambient pressure is not too high, so there is no problem on the surface or even in water depth of less than 3500 ft. In these circumstances, Boyles Law, $P_1V_1 = P_2V_2$, along with a safety factor of 1.5 is used to calculate an accumulator's usable fluid. Dividing the volume of hydraulic fluid to close and open one annular-type preventer and all ram-type preventers from a full-open position against atmospheric wellbore pressure by the volume of usable fluid of each accumulator, the number of accumulators can be determined. But there is another story when accumulators are to be located in deeper waters. The subsea accumulator capacity calculations should compensate for the hydrostatic pressure gradient at the rate of 0.445 psi/ft of water depth.[2] In these circumstances, nitrogen used in accumulators does not behave like an ideal gas. It behaves like a non-ideal gas and we have to incorporate Z-factor in our calculations and also consider the process of expansion of nitrogen as an adiabatic process. This model that is the most accurate description of the expansion of the gas[3] calculates the pressure that the accumulator initially contains after a discharge. This method even gives a smaller usable fluid volume than what you see in Fig. 1-1 for deeper waters. In ultra-deepwater drilling, the usable fluid of accumulators is much less than that of the same accumulators on the surface or in relatively shallow waters. So, we have to provide our hydraulic control system with a large number of accumulators that costs too much, and even worse, it would be very difficult to use this kind of accumulator in water depth of more than about 12,000 ft where today's investigators in the petroleum industry are trying to reach.

Since the volumetric efficiency of gas charged accumulators is so low in ultra-deep water and no replacement for gas has been found yet, some investigators are trying to find a way to transfer all the BOP equipment to the surface.

In most cases, a properly sized rigid conduit to conduct power fluid from accumulators mounted on the surface to the BOP stack is capable of providing sufficient flow rates to operate BOP functions within API mandated limits, even in extreme water depths.[4] However, certain BOP control systems must have dedicated sources of stored hydraulic fluid located on the BOP stack.[4]

Replacing conventional accumulators by another kind of accumulator whose functionality is not affected by the hydrostatic pressure, may provide a solution.

Spring-loaded accumulators and weighted accumulators are discussed in this thesis as two possible replacements for gas charged accumulators. Lifted weight and deflected springs can store energy like compressed gas. The work done on a deflected spring is stored in it and can be released at the time required. This energy can be used to run the piston inside a cylinder like the compressed gas in the piston type of gas charged accumulator. There is the same story about lifted weight; the work of the weight of the body in vertical displacement is stored as potential energy and can be recovered to run the piston against the hydraulic power in a cylinder.

CHAPTER II

ACCUMULATORS

2.1 Different Kinds of Accumulators

Accumulator bottles are containers that store hydraulic fluid under pressure for use in effecting BOP closure. Through use of compressed nitrogen gas, these containers store energy that can be used to decrease BOP function response time, and to serve as a backup source of hydraulic power in case of pump failure. There are three types of accumulators in common usage, bladder, piston, and diaphragm. Identical in their operating principle, these three kinds of accumulators use different mechanism to separate gas from the hydraulic fluid. It is the difference - and the resulting performance characteristics - which determine their suitability for different applications.

Piston accumulators. Piston accumulators consist of a cylindrical body, sealed by a gas cap and a charging valve at the gas end, and a hydraulic cap at the hydraulic end. A lightweight piston separates the gas side of the accumulator from the hydraulic side. The gas side is charged with nitrogen to a predetermined pressure. Changes in System pressure cause the piston to rise and fall, allowing fluid to enter or forcing it to be discharged from the accumulator body. **Fig. 2.1** shows the different parts of a typical piston accumulator.

This kind of accumulator is made in different capacities; for example, Hydac manufactures 50-gallon and smaller accumulators with the maximum working pressure of 5,000 psi, 80-gallon effective gas volume and smaller with the maximum working pressure of 3,000 psi. These accumulators are manufactured in different dimensions, sometimes they are 170 in. long and 14 in. diameter.

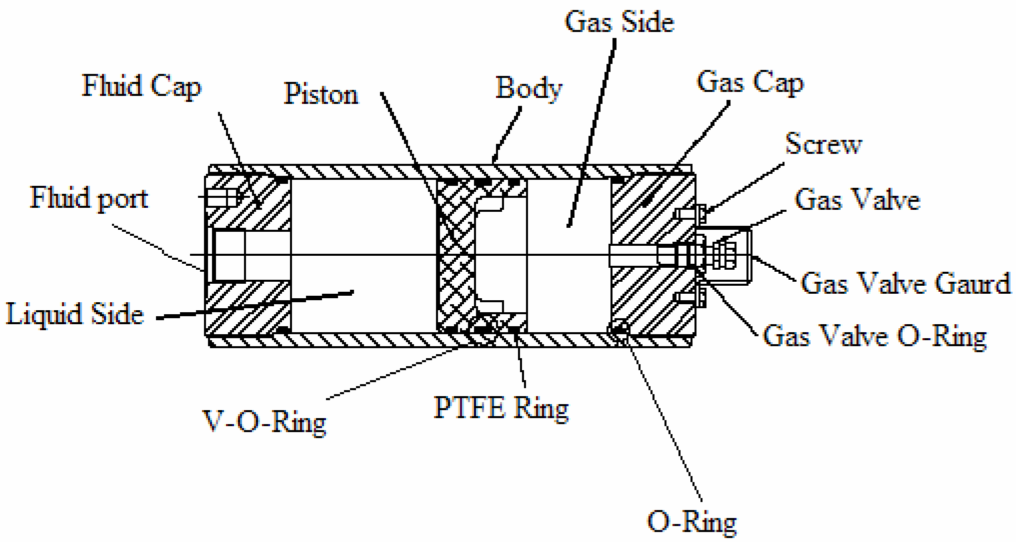


Fig. 2.1- A typical piston accumulator. (After Parker[5])

Extremely high flow rates, high/low temperature tolerance, high compression ratios, withstand external forces, unlimited sizes, working well with gas bottles, and fully serviceable can be named among advantages of this kind of accumulator.

Diaphragm accumulators. Diaphragm accumulators contain a one piece molded diaphragm which is mechanically sealed to the high strength metal shell. In this kind of accumulator, the flexible diaphragm provides gas and fluid separation. A button molded to the bottom of the diaphragm prevents the diaphragm from being extruded out the hydraulic port. The non-repairable electron-beam welded construction reduces size, weight, and ultimately cost. The diaphragm accumulator is charged with a dry gas such as nitrogen, to a set precharge pressure determined by the system requirements. As the system pressure fluctuates, the diaphragm expands and contracts to discharge fluid from, or allow fluid into, the accumulator shell. In **Fig. 2.2**, a typical diaphragm accumulator is shown in detail.

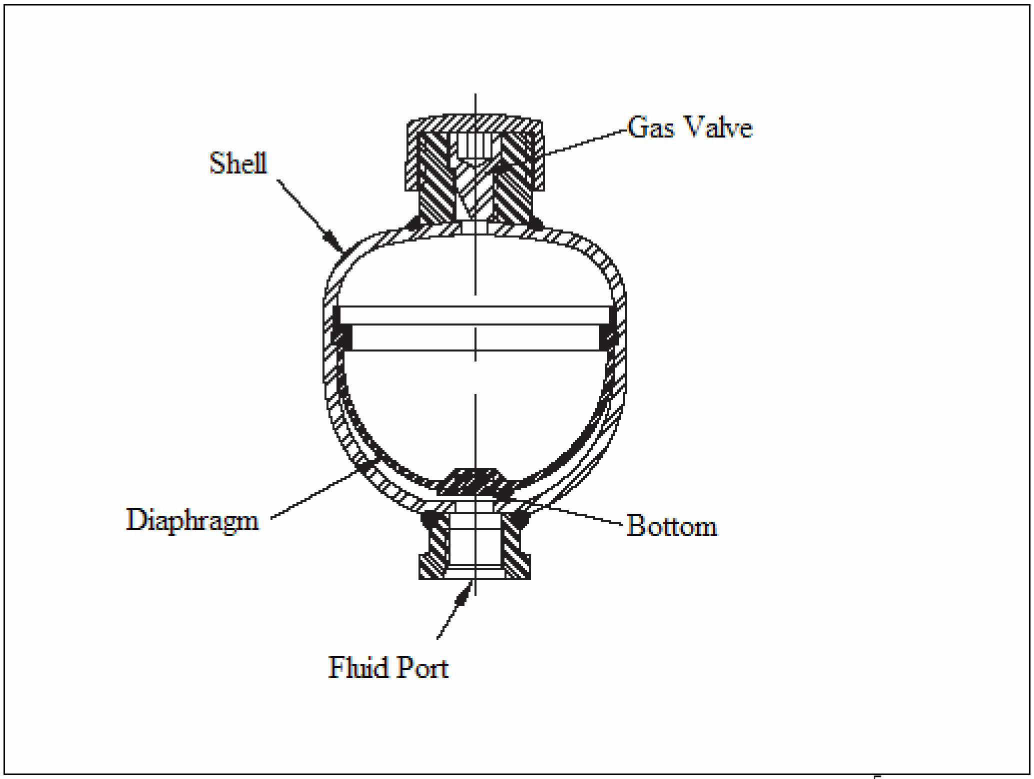


Fig. 2.2- A typical diaphragm accumulator. (After Parker[5])

These kinds of accumulators are compact and also lightweight. They are simple and can be built to be cost effective. Response time in diaphragm accumulators is shorter than in piston accumulators. Diaphragm accumulators are not installed in hydraulic control system circuit for subsea BOP stacks due to their low capacity and low maximum operating pressure (3600 psi).[5] They usually can hold a volume of fluid less than 1 gallon.

Bladder accumulators. In this kind of accumulator a non-pleated, flexible rubber bladder is housed within a steel shell. The open end of the bladder is attached to the precharging valve at the gas end of the shell. A poppet valve, normally held open by spring force, controls flow rate, with excessive flow causing the poppet to close permanently.[5] Just like the other kinds of accumulator, the bladder accumulator is charged with nitrogen. As fluid is pumped into the fluid end, the bladder with nitrogen in it contracts to allow fluid into the accumulator shell, and this high pressure bladder expands to discharge fluid from the accumulator bottle. A typical bladder accumulator looks like what you can see in **Fig. 2.3.**

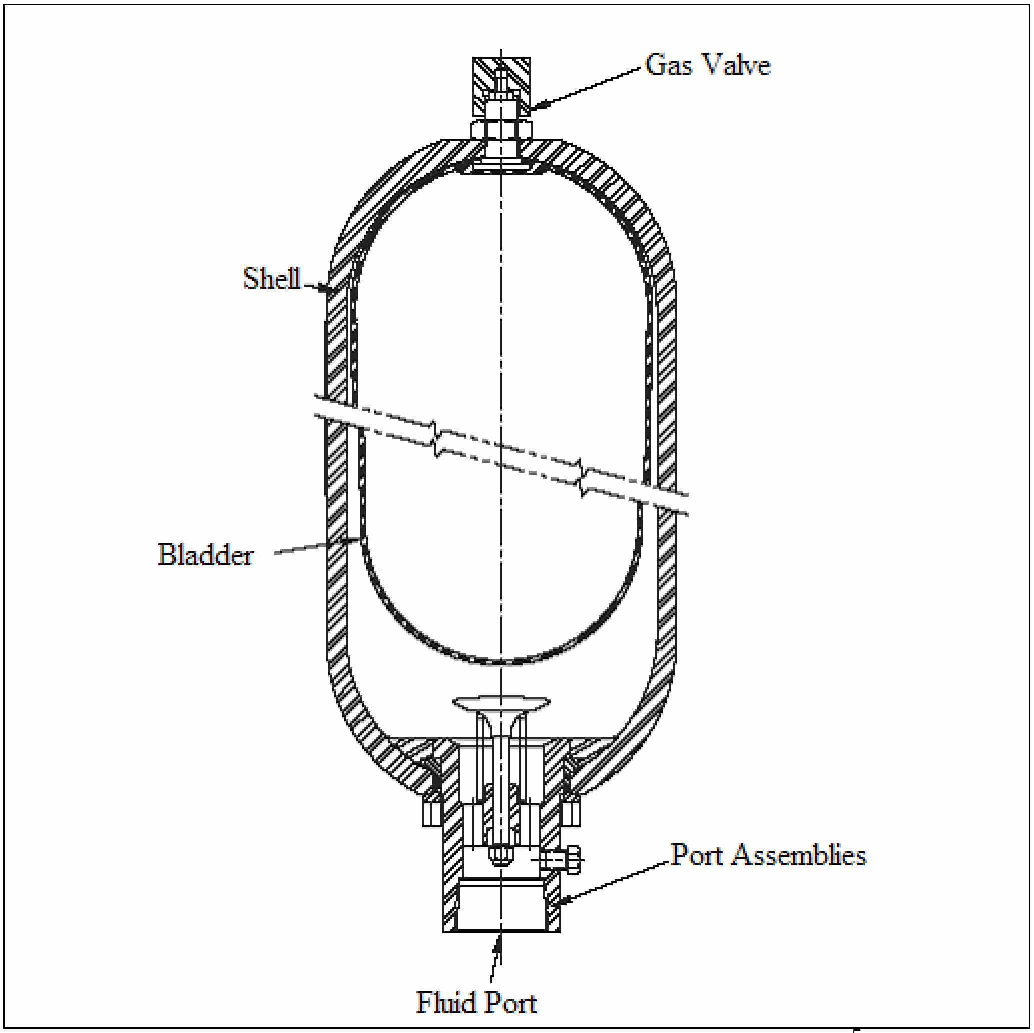


Fig. 2.3- A typical bladder accumulator. (After Parker[5])

The most common kind of accumulator used in the petroleum industry in BOP control systems for drilling purposes is the bladder accumulator. Bladder accumulators are generally preferred for applications where rapid cycling, high rate fluid communication and fast response times are required.[5] There is no static friction to be overcome as with a piston seal, and there is no piston mass to be accelerated, so response time for such accumulators are very short and less than 25ms. Bladder accumulators are also more resistant to damage cause by contamination of the hydraulic fluid than piston types.

These accumulators are available from 1-gallon to 15-gallon sizes with different maximum working pressures of 3,000 and 5,000 psi.

2.2 Operation

Accumulators do nothing unless otherwise charged. An empty accumulator has nothing in it and can do nothing; neither gas nor hydraulic sides are pressurized. An accumulator should be precharged with nitrogen to a predetermined pressure before it is charged with hydraulic fluid. The minimum precharge pressure for BOP stack installation for a 3,000 psi working pressure accumulator unit should be 1,000 psi.[2] The minimum precharge pressure for a 5,000 psi working pressure accumulator unit should be 1,500 psi.[2] A precharged accumulator is ready to be charged with hydraulic fluid up to its rated working pressure which is the maximum internal pressure that accumulators are designed to hold. The accumulator is filled with hydraulic fluid to its design capacity. Now, the accumulator is a source of energy and the pressurized fluid inside the accumulator is ready to flow into the system at the time you need. Pressure available in the gas side forces fluid from the accumulator into the system. The accumulator is discharged until the minimum system pressure is reached. API 16D mandates that after closing and opening one annular preventer and all ram-type preventers, the remaining pressure shall be 200 psi or more above the minimum recommended precharge pressure, which is called minimum working pressure.[2] When the minimum working pressure is reached, a small volume of fluid will remain in the accumulator. At this time it has discharged its design maximum volume of fluid into the system. **Fig. 2.4** shows operating stages of bladder, piston and diaphragm accumulators.

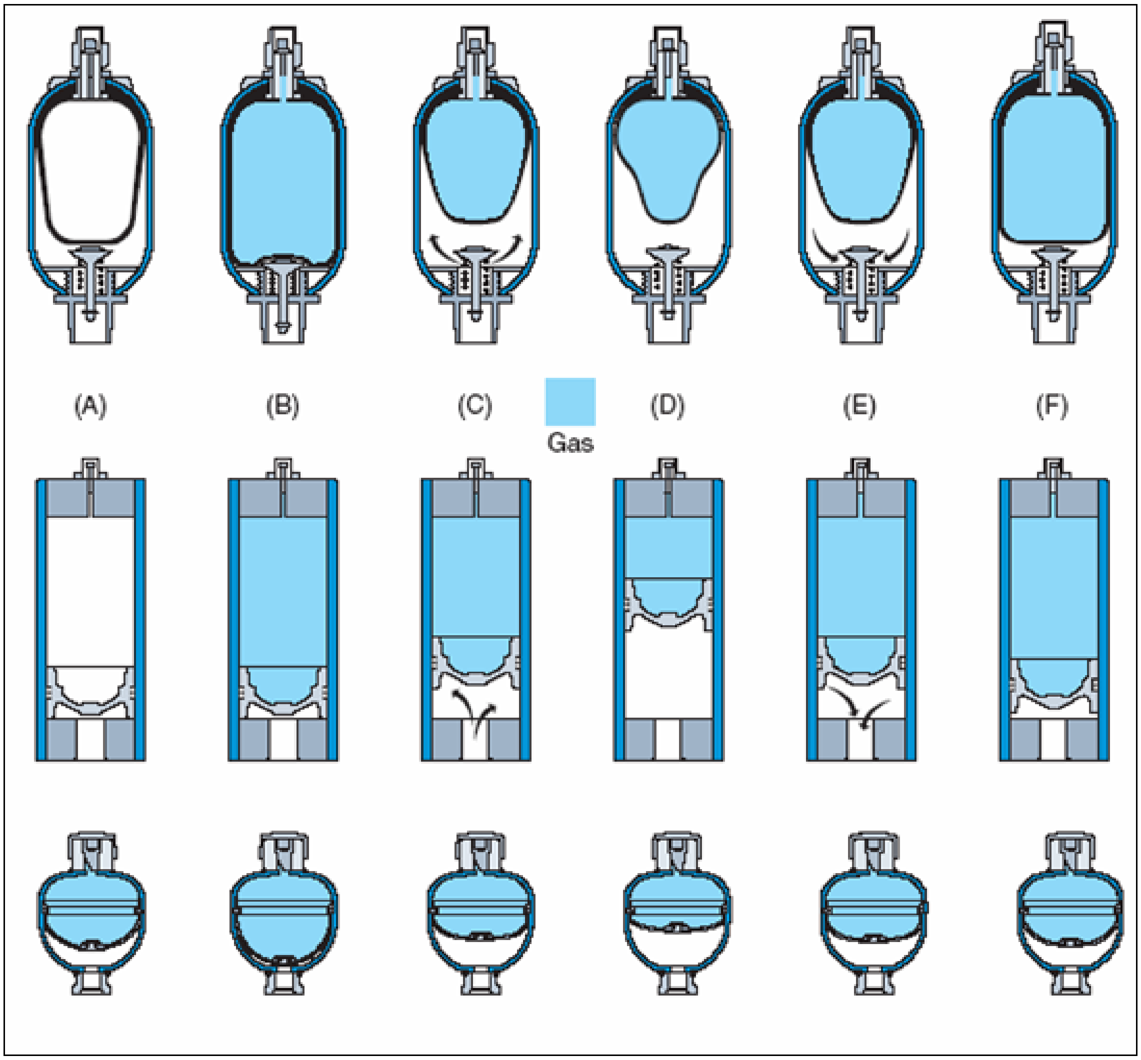


Fig. 2.4- Operating stages of bladder, piston and diaphragm accumulators. A) empty accumulator, B)precharged accumulator, C)being charged, D)charged state, E)discharging, and F)discharged.[5]

2.3 Precharge Pressure

Precharging process is accurately filling the gas side of an accumulator with a dry, inert gas such as nitrogen, before admitting fluid to the hydraulic side. Although the definition of the precharge pressure implies that it is possible to use another inert gas rather than nitrogen, the only gas recommended by API is nitrogen.[2]

It is important to precharge an accumulator to the correct specified pressure. Precharge pressure determines the volume of fluid retained in the accumulator at minimum system pressure. As mentioned before, the minimum precharge pressure recommended by the API for a 3,000 psi working pressure accumulator unit should be 1,000 psi and for a 5,000 psi working pressure accumulator unit should be 1,500 psi. The manufacturer of the accumulator also recommends a specific precharge pressure for that accumulator. A bladder accumulator is typically precharged to 90% of minimum system pressure, and a piston accumulator to 95% of minimum system pressure at the system operating temperature.[5] There is no contradiction between precharge pressure recommended by API and precharge pressure recommended by the manufacturer when accumulators are made for drilling purposes.

It is really important to precharge an accumulator to the correct accumulator precharge. Excessive precharge pressure or a reduction in the minimum system pressure without a corresponding reduction in precharge pressure may cause operating problems or damage to accumulators.[5] Excessive precharge pressure in a piston accumulator will cause the piston to travel too close to the hydraulic end cap. The piston could bottom at minimum pressure, reducing output and eventually damaging the piston and piston seal. The excessive precharge pressure can damage bladder accumulators, too. An excessive precharge pressure in bladder accumulator can drive the bladder into the poppet assembly. This could cause fatigue failure of the poppet spring assembly, or even a pinched and cut bladder, should it become trapped beneath the poppet as it forced closed. Excessive precharge pressure is the most common cause of bladder failure.[5] Excessive low precharge pressure or an increase in system pressure without a corresponding increase in precharge pressure can also cause operating problems and subsequent accumulator damage.[5] With no precharge in a piston accumulator, the piston will be driven into the gas end cap and will often remain there. For a bladder accumulator, too low or no precharge pressure can have rapid and severe consequences. As you can see in **Fig. 2.5** the bladder will be crushed into the top of the shell and can extrude into the gas stem and be punctured.

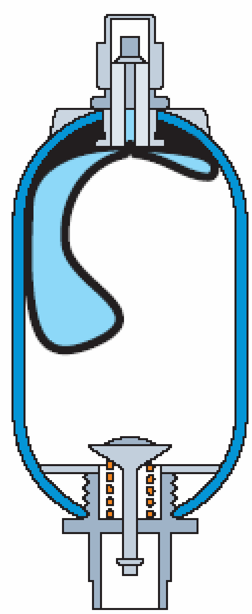


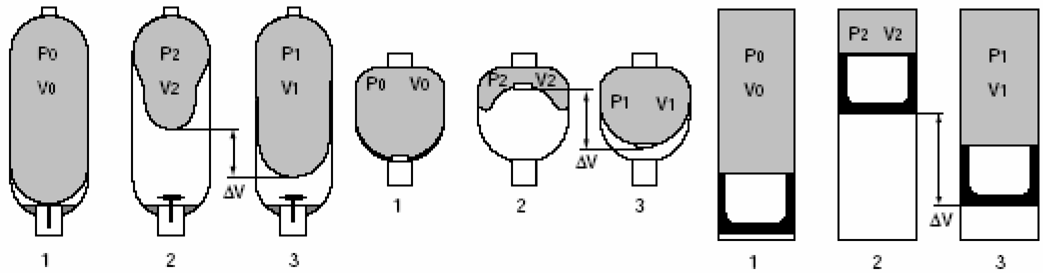
Fig. 2.5- Fluid has entered an un-precharged bladder accumulator. It has forced the bladder into the gas stem.[5]

2.4 Accumulator Fluid Capacity

The fluid capacity of a hydro-pneumatic accumulator is defined as the total fluid (liquid and gas) that the accumulator will hold. It must exclude the volume of bladder (or piston) and the volume of internal valve and other internal objects. For example the actual fluid capacity of nominal 15-gallon bladder accumulator may be 13.7 gallons.

2.5 Usable Fluid

The hydraulic fluid volume recoverable from the accumulator between the maximum operating pressure and the minimum operating pressure is defined as its usable fluid. **Fig. 2.6** gives you more understanding of this concept. The usable fluid for a hydro-pneumatic accumulator is not constant and changes depending the maximum and minimum operating pressure and precharge pressure.



P0 = Precharge Pressure
P1 = Minimum Working Pressure
P2 = Maximum Working Pressure
V0 = Effective Gas Volume of Accumulator
V1= Gas Volume at P1
V2= Gas Volume at P2
ΔV = Usable Fluid

Fig. 2.6- Usable fluid of three kinds of accumulators. (After Hydac[6])

2.6 Accumulator Selection

It is obvious that the volume of power fluid you need is an important factor to select an accumulator. System pressure also should be considered. Maximum working pressure and minimum working pressure must be known. Temperature affects the behavior of the gas, so in which temperature the accumulator should operate is very important. Installation space and position are factors that should be taken under consideration. For BOP purposes, accumulators are selected to be positioned vertically. Materials which the accumulator is made of should also be resistant against possible corrosion in offshore drilling, too. Over all, bladder accumulators are usually selected for installation in subsea BOP control systems. Fast response time and acceptable cost are some features of this kind of accumulator.

2.7 Accumulator in Subsea BOP Control System Circuit

Blowout preventers are designed and connected vertically to the casinghead by hydraulic connector to shut in a well under pressure so that formation fluids that have moved into the wellbore can be circulated out while continuous control of the well is maintained. More than one type of BOP is used and, for redundancy, two or more preventers, of the same type are used (**Fig. 2.7**). These will be stacked together and the assembly is referred to as a BOP stack, or simply the stack (**Fig. 2.8**). Another hydraulic connector is also used to connect the top of the stack to the riser. All the rams, preventers, and valves installed on the stack can be actuated hydraulically.

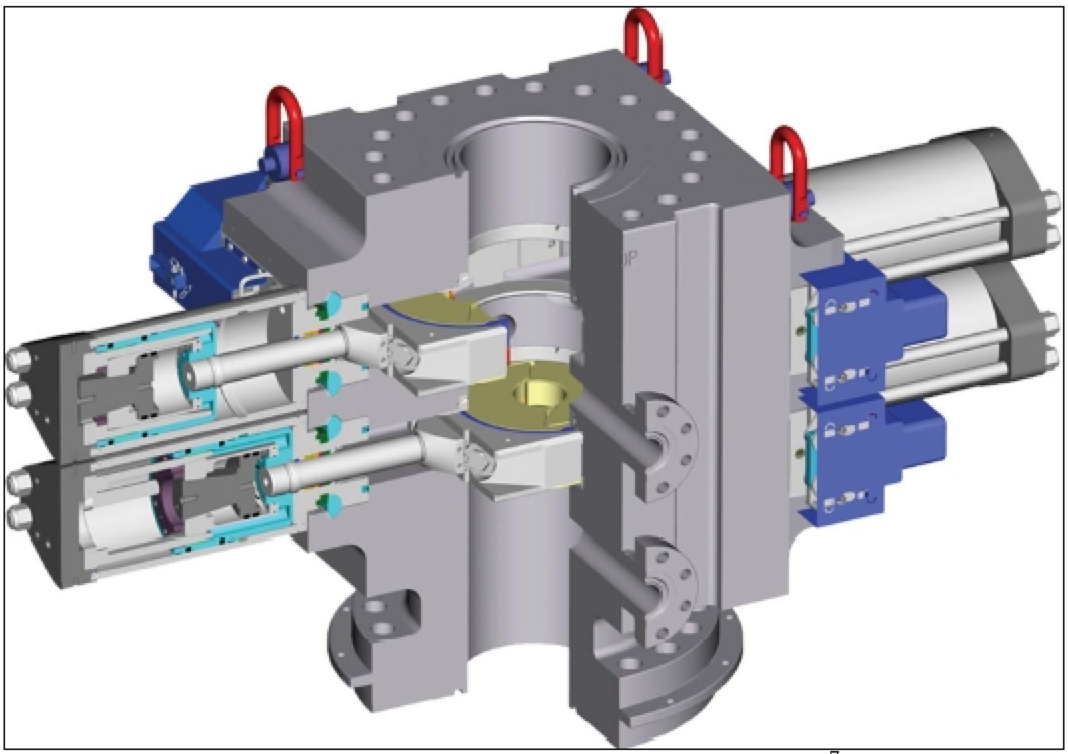


Fig. 2.7- Shaffer NXT-type blowout preventer.[7]

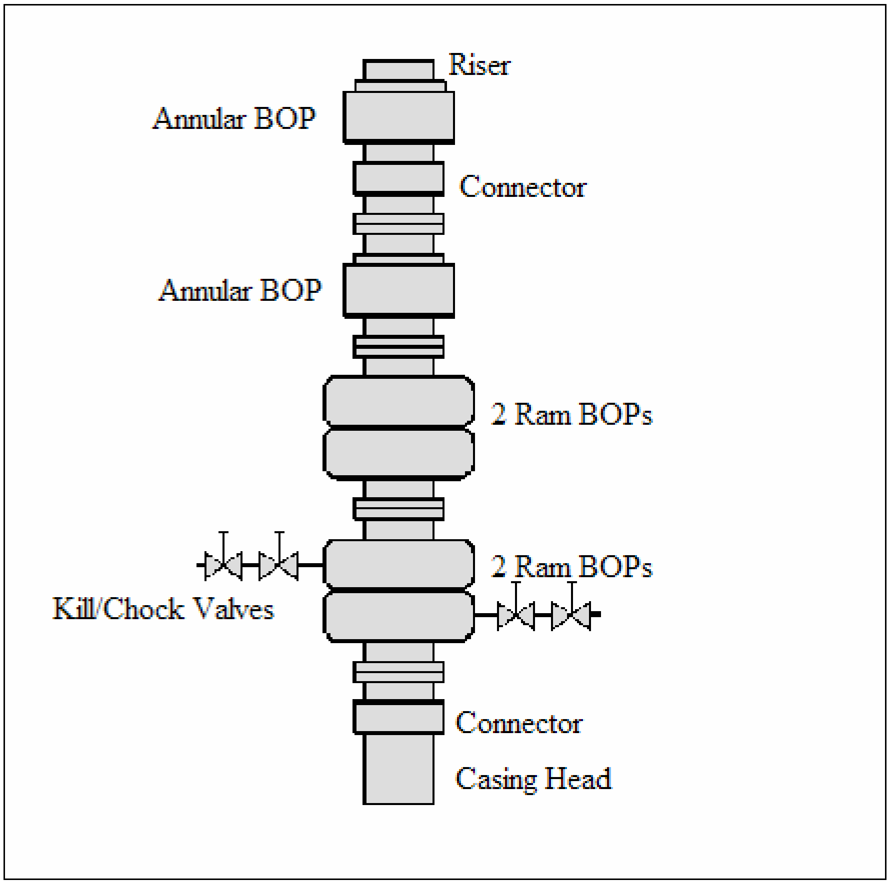


Fig. 2.8- Schematic of the BOP stack.

Pipe rams are used to close off the annular space outside the drillpipe. Using the pipe ram, an entire string of drillpipe and collars may be hung off from a tool joint landed on a ram. Conventionally, three different sizes of pipe rams are used to seal off around the pipe inside the well. A pipe ram may look like what is shown in **Fig. 2.9**. A fourth ram, a shear ram, is also incorporated into the stack. A type of shear ram manufactured by Varco Company is shown in **Fig. 2.10**. A Shear ram is used to seal off the open hole and to shear the drillpipe when necessary. Shearing the pipe is of course one of the last resorts in an emergency situation.

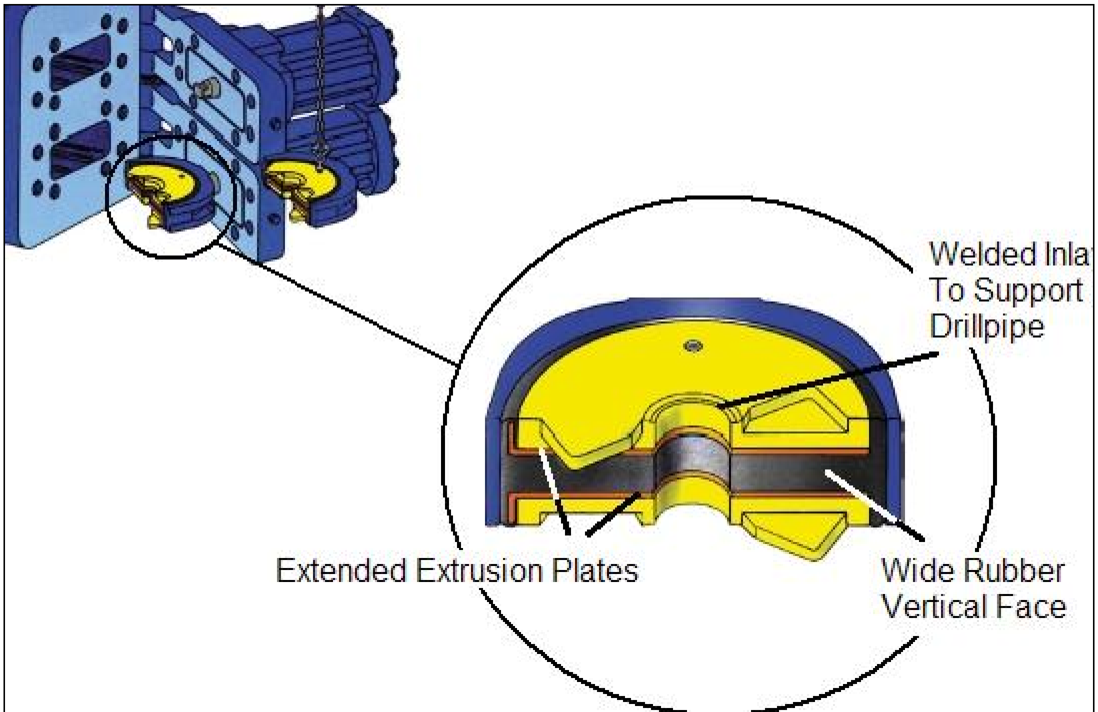


Fig. 2.9- A pipe ram.[8]

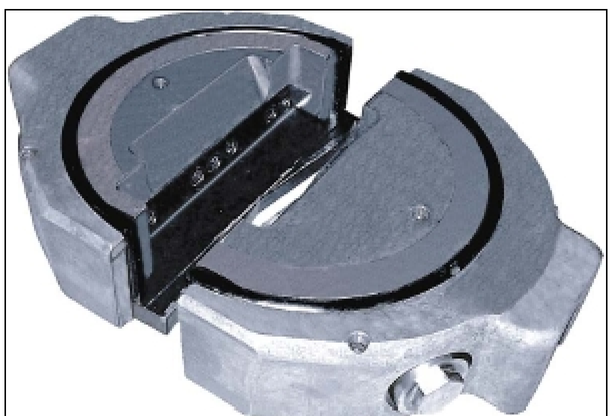


Fig. 2.10- A kind of shear ram.[9]

Annular preventers are another part of the BOP stack. Annular preventers are comprised of specially designed, reinforced rubber elements that can seal around any tubular or nearly tubular objects that will go through the BOPs. They will also seal over the open hole, and can pass drillpipe joints without sever damage to the sealing elements. They are used for stripping drillpipe into the well under pressure and for shutting the well around the pipe when the pipe rams can not be used. In particular, annular preventers are used for initial shut-in during a kick and to hold the wellbore pressure until a drillpipe tool joint is located and hung off on a pipe ram. Frequently, two annular preventers will be used. One of these preventers will normally be located above the upper hydraulic connector so that it can be retrieved with the riser. A typical annular preventer is shown in **Fig. 2.11** in detail.

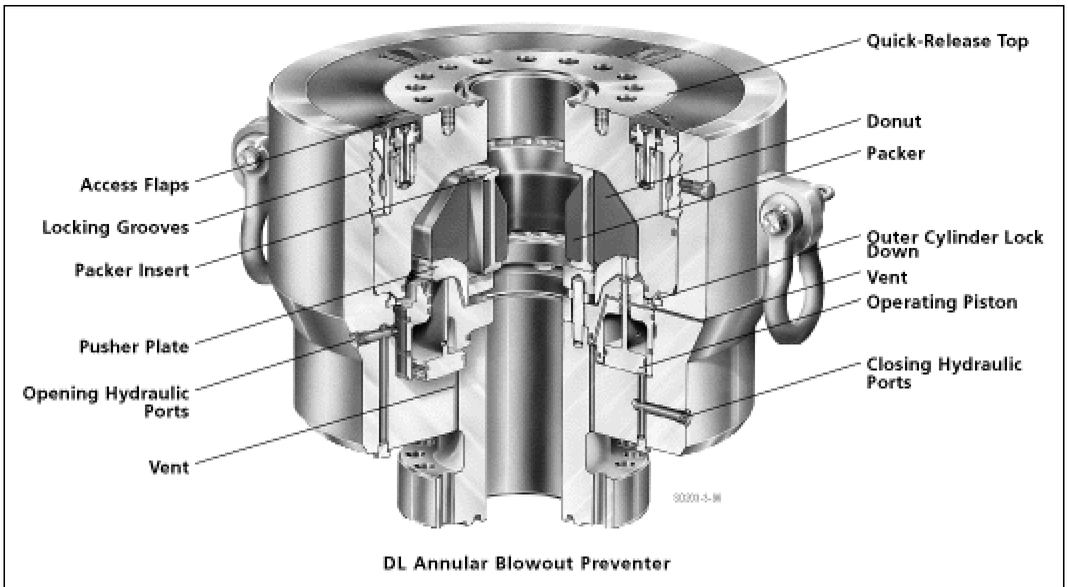


Fig 2.11- Annular blowout preventer.[10]

Kill and choke valves are also part of the BOP stack. Kill and choke valves are the subsea shut off for the high pressure kill/choke lines that run from the BOP to the choke manifold on the rig. K&C valves are hydraulically controlled from the surface and are designed to close by spring action when opening pressure is released. Two valves in each line should always be used for redundancy.

All those elements incorporated into the BOP stack are actuated hydraulically, so we need a control system to control them on the stack at the time required. Control systems for the BOPs are of necessity highly efficient hydraulic systems. The objective is to operate any of the functions on the stack in as short time as possible. This requires high flow rates for the large volumes needed to operate the major functions such as the rams and annular preventers. For example, eight seconds is an acceptable closing time for a ram. Volumes required for closing preventers varies from about five gallons for small rams to over fifty gallons for a large annular preventer. The fluid used to operate the functions on the stack is mixed, pressurized and delivered from the hydraulic unit. The fluid is passed through a hose bundle to a subsea control pod. Fluid flows from the active pod and is diverted by shuttle valve to the function on the BOP stack. Accumulators are located in this hydraulic circuit to store energy. Usually more than one accumulator is used to store hydraulic fluid under pressure. All the accumulators are manifolded together in the bank of accumulators. The bank of accumulators and some other equipment including pressure pumps to bring the control fluid up to accumulator pressure, rotary valves, air pistons to operate the rotary valves from a remote location, and a selector valve to direct the flow of either power or pilot fluid to the active pod and isolate the inactive pod comprise the hydraulic unit. Of course an atmospheric storage tank for control fluid is required as a part of the hydraulic unit. In **Fig. 2.12**, you can see the position of accumulators in the circuit of a BOP control system in connection with a BOP stack.

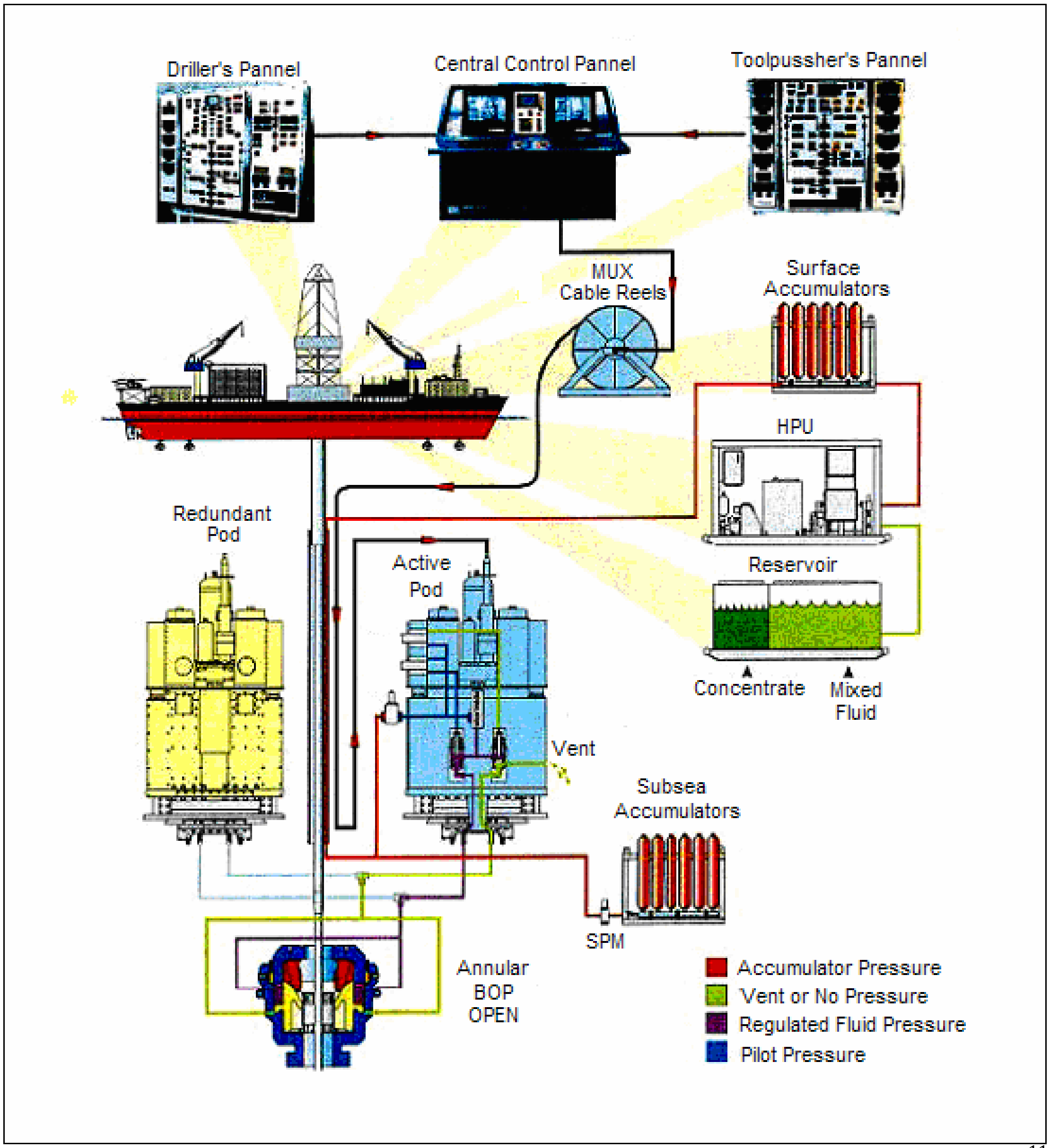


Fig 2.12- Schematic of a hydraulic control system in connection with the BOP stack.[11]

2.8 Accumulator Capacity Requirements

The BOP control system shall have a minimum stored hydraulic fluid volume, with pumps inoperative, to satisfy the greater of the two following requirements:

1. Close from a full open position at wellbore pressure, all of the BOP's in the BOP stack, plus 50% reserves.[12]

2. The pressure of the remaining stored accumulator volume after closing all of the BOP's shall exceed the minimum calculated (using the BOP closing ratio) operating pressure required to close any ram BOP (excluding the shear rams) at the maximum rated wellbore pressure of the stack.[12]

Table 2.1 is an example of calculation of total fluid volume requirements for surface accumulators.

As you can see in Table 2.1, precharged pressure is considered to be 1,000 psig (1,015 psia). Accumulator gas precharge (gas pressure with the liquid side open to the atmosphere) of 1,000 psig is generally accepted, because it is at least 40% higher than the minimum pressure needed to operate any function on the stack including the pilot valves. Fluid storage pressure is conventionally 3,000 psig maximum. It is the maximum pressure rating for off-the-shelf equipment, and is an economic upper limit. These limits may change in deepwater drilling where the efficiency of the 3,000 psi system decreases.[13] We discuss this later in this thesis.

For subsea accumulators, we divide the value 265.5 by the usable fluid fraction of the accumulators at the depth of interest. For example for a usable fluid fraction of 0.3, total fluid capacity required should be 885 gallons not 400 gallon. In the next chapter we discuss how to calculate usable fluid fraction for subsea accumulators.

To calculate the number of accumulators, we just divide the total fluid capacity required by the actual volume of the accumulators in use.

Table 2.1- Fluid volume requirements for surface accumulators.[13]

Nominal stack size 16¾, pressure rating 5,000 psi, K&C Valve size 4 in.				
Function	**Make**	**No. of Units**	**Volume per Function (gal)**	**Volume Requirements (gal)**
Ram close	Cameron	4	10.6	42.4
Ram open	Cameron	4	9.8	39.2
Annular close	Hydril	1	51.1	51.1
Annular open	Hydril	1	33.8	33.8
K&C valves open	McEvoy	6	1.3	7.8
K&C valves close	McEvoy	3	0.9	2.7

Liquid volume to open and close functions	177.0 gal
Safety Factor	X 1.5
Liquid volume required	265.5 gal
Precharge Pressure (P_n) = 1,015 psia	
Working Pressure (P) = 3,015 psia	
Volume Factor = $1 - P_n / P = 0.663$	divided by 0.663
Total fluid capacity required	400 gal

CHAPTER III

LIMITATION OF GAS CHARGED ACCUMULATORS

3.1 Offshore Drilling

Technology and design and use of drilling equipment on land were well established at the advent of floating drilling; however it became apparent that when BOPs are located on the seafloor, considerable changes on the equipment were required. Major changes in the equipment for subsea use are:

1. The size of the BOPs is increased.[13]
2. The philosophy of stack arrangement, especially the location of the kill and choke lines has been changed.[13]
3. Pressure drop in the long choke line(s) influences well control procedures.[13]
4. To avoid the pressure drop in return lines, the hydraulic fluid is vented to the sea. This requires a fluid that is non-polluting as well as non-corrosive, yet has low viscosity, is a good lubricant and can mix with water of high mineral content.[13]
5. External hydraulic pressure at the ocean floor must be considered.[13]
6. Hydraulics have become more important in reaction times, because the longer flow lines increase the pressure drop, while the larger BOPs require more fluid to operate than their land counterparts.[13]

3.2 Response Time and Local Storage

As you can see in the list above, reaction time is one of the concerns of floating drilling projects. The elapsed time between activation of a function at the control panel on the surface and complete operation of the function on the stack is called response time. Industry standards, API and MMS,[11] for BOPs require ram BOPs to close within 45 seconds and annular BOPs in 60 seconds; however it is an industry goal to close these critical functions as fast as possible.

Measurement of closing response time begins at pushing the button or turning the control valve handle to operate the function and ends when the BOP or valve is closed. Two factors affect BOP control response time:

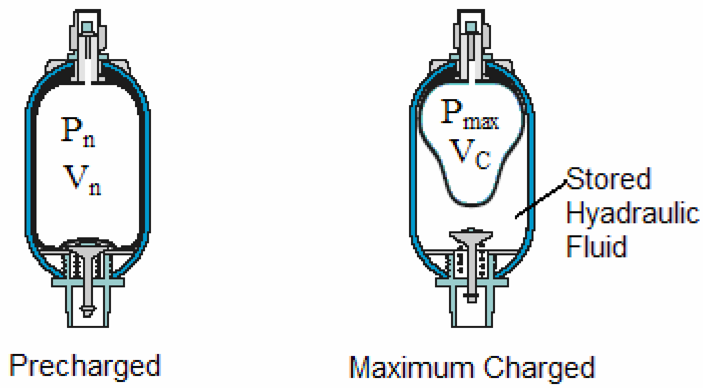
1. Signal time from surface to actuate the pod mounted directional control valve.
2. Hydraulic flow time from the main supply through the pod mounted directional control valve to the BOP function, for example rams-close.[4]

The sum of signal time and hydraulic flow time should meet industry regulations and standards. Subsea accumulators have no influence on the signal time. The speed of BOP operation depends largely on how fast the hydraulic power can flow from the main supply through the directional control valve and on to the BOP function. The shorter the conduit from the main supply to the BOP stack, the shorter the hydraulic flow time. So, we should install the main accumulators as close as possible to the stack to reduce hydraulic response times. These accumulators are also used in subsea production control systems to provide local storage that allows smaller line sizes in control umbilicals.[1] Subsea accumulator as a local hydraulic power supply becomes even more important in case of interruption of surface communication.

Mounting accumulators at the sea floor reduces response time enough to meet the governing regulations and standards, but a new problem arises. Volumetric efficiency of these gas charged accumulators decreases when they encounter the high hydrostatic pressure at the sea floor. To make this matter more clear, let's get into calculation of usable fluid. In the following section we discuss the concept of usable fluid and its calculation method. Volumetric efficiency of an accumulator is defined as the volume of usable fluid as a percentage of its actual fluid capacity.

3.3 Usable Fluid Calculation

Imagine an accumulator precharged with nitrogen to the pressure P_n. At this time, there is no liquid in the liquid side of the accumulator. As we have mentioned in chapter II, accumulators in the BOP control system should be charged with a volume of proper power fluid to the maximum working pressure; let's call it P_{max}. **Fig. 3.1** demonstrates how much fluid we need to pump into the accumulator bottle to get the maximum working pressure. Since we are using a gas equation of state, all pressures should be measured in absolute pressure units.



P_n = Precharge Pressure,
V_n = Volume of gas at the precharge pressure,
P_{max} = Maximum Working Pressure,
V_c = Volume of gas when the accumulator is charged.

Fig. 3.1- Stored hydraulic fluid of accumulator.

To calculate the volume of fluid pumped into the accumulator bottle, we need to know the actual fluid capacity of the accumulator, and also calculate the volume of gas at the maximum working pressure. Subtracting volume of gas from actual fluid capacity of the accumulator gives us volume of fluid required to pump into the accumulator bottle. We use Boyles Law, $P_1V_1 = P_2V_2$, to calculate volume of gas at the maximum operating pressure. In this model we consider nitrogen as an ideal gas, and also, we do not take into account the temperature changes that occur temporarily in the accumulator while charging the accumulator. So, for the accumulator in Fig. 3.1 we can write:

$$P_n V_n = P_{\max} V_c \text{ , } \quad (3.1)$$

Now, we solve the Eq. 3.1 for V_c:

$$V_c = \frac{P_n V_n}{P_{\max}} \text{ , } \quad (3.2)$$

Then, subtracting V_c from the actual fluid capacity of the accumulator, V_{ac}, gives the volume of power fluid pumped into the bottle, V_p:

$$V_p = V_{ac} - V_c \text{ , } \quad (3.3)$$

Replacing V_c by Eq. 3.2 in Eq. 3.3 we get:

$$V_p = V_{ac} - \frac{P_n V_n}{P_{\max}} \text{ , } \quad (3.4)$$

As it can be seen in Fig. 3.1, actual fluid capacity of an accumulator can be assumed to be equal to the volume of gas at the precharge pressure, $V_{ac} = V_n$. Replacing V_n by V_{ac} in Eq. 3.4 and rearranging it, we will have:

$$V_p = (1 - \frac{P_n}{P_{\max}}) V_{ac} \text{ , } \quad (3.5)$$

This V_p is called stored hydraulic fluid of the accumulator. But the whole volume of stored hydraulic fluid pumped into the accumulator is not recoverable. After discharging, we should have 200 psi or more above precharge pressure in the system. This minimum pressure is prescribed by the manufacturer.[2] Now let's calculate the volume of fluid left inside the accumulator bottle, V_{left}, after discharging the accumulator to the minimum working pressure. As you can see in **Fig. 3.2**, the volume of liquid left inside the accumulator is:

$$V_{left} = V_{ac} - V_d \, , \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.6)$$

where V_d is volume of gas at the minimum operating pressure when the accumulator is discharged. V_d can be calculated in the same way as V_c:

$$V_d = \frac{P_n V_n}{P_{\min}} \, , \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.7)$$

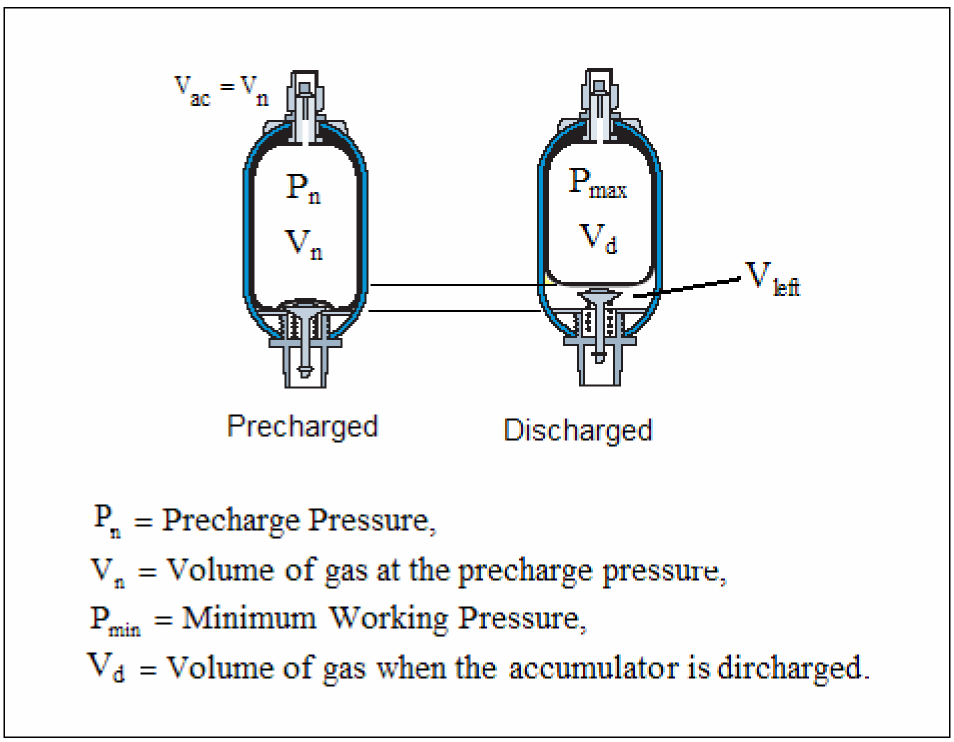


Fig. 3.2- Volume of liquid left inside the accumulator.

Recalling $V_n = V_{ac}$ and substituting Eq. 3.7 in Eq 3.6 gives us:

$$V_{left} = (1 - \frac{P_n}{P_{min}})V_{ac}\,, \quad \text{(3.8)}$$

As you can see, the whole volume of fluid pumped inside the accumulator is not recoverable and there is a volume of fluid left inside the bottle. The volume of fluid recoverable between maximum working pressure and minimum working pressure, which is less than stored hydraulic fluid, is called usable fluid and can be calculated by Eq. 3.9:

$$V_U = V_p - V_{left}\,, \quad \text{(3.9)}$$

where V_U is usable fluid.

Now, we can generate a general formula for accumulators' usable fluid (Eq. 3.10). We can put Eq. 3.5 and Eq. 3.8 in the Eq. 3.9 to get the Eq. 3.10:

$$V_U = \left(\frac{P_n}{P_{min}} - \frac{P_n}{P_{max}}\right)V_{ac}\,, \quad \text{(3.10)}$$

The parenthesis in Eq. 3.10 is called usable fluid fraction. Traditionally subsea accumulator capacities have been calculated based on this method; isothermal model.[1] The name isothermal indicates that this method does not take into account temperature changes that occur temporarily in the accumulator while charging and discharging. This method has given adequate accuracy in surface and relatively shallow subsea systems, partially due to the generous safety margins that are built into mandated capacities.[1] For operation in 3,000 ft of water on 3,000 psi control systems, using Boyles Law to calculate usable subsea accumulator volumes results in overstating capacity slightly more than 15% .[3] This has not been a problem because of design factor of 1.5.[4] This method is accepted by API 16D.

Accumulator system pressure is higher at any water depth than at the surface in order to compensate for hydrostatic head. The subsea accumulator bottle capacity calculations should compensate for hydrostatic pressure gradient at the rate of 0.445 psi/ft of water depth.[2] There is the same story about precharge pressure; the precharge pressure should also compensate for the water depth. So, we can rewrite Eq. 3.10 as Eq. 3.11 to calculate accumulator usable fluid installed in relatively shallow water, less than 3500 ft.

$$V_U = \left(\frac{P_n + 0.445 D_w}{P_{\min} + 0.445 D_w} - \frac{P_n + 0.445 D_w}{P_{\max} + 0.445 D_w} \right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.11)$$

Where D_w is water depth measured in ft.

3.4 Non-ideal Gas Behavior

Nitrogen does not behave like an ideal gas at higher pressure; more than 3,500 psi is accepted. The two major reasons why nitrogen does not act ideally and does not obey Boyles Law at higher pressures are the physical sizes of the molecules and intermolecular forces. Because molecules, even as gases, have size, they take up space. Under low pressure, the volume occupied by gas molecules is rather insignificant relative to the container volume. However, at high pressures, molecules take up a higher percentage of the confined volume and the likelihood of interaction with other molecules of the gas increases. It is these intermolecular forces that lead to the departure from ideal gas behavior.

Similarly, high pressure forces molecules to be closer together. Gas molecules interact with each other if they are in closer proximity by what are termed Van Der Waals forces. These forces result in electrostatic repulsion caused by the subatomic differences in charge of the individual molecules found in very close proximity.

Both size and the presence of intermolecular forces result in gas molecules not compressing gas as much as would be theoretically predicted by the ideal gas law.

3.5 Deepwater Conditions

Today, it is not uncommon to drill in water depths in excess of 7,000 ft. As we have mentioned before, for the most part, the long-standing practice of installing accumulators on subsea BOP stacks in an effort to decrease BOP response times has persisted. However, accumulator system pressures are substantially higher in these deeper waters. In addition, some BOP control system design pressures have increased, further increasing the pressure inside subsea accumulators.[4] Wellbore pressure also increases due to pressure associated with a kick. As you can see in **Fig. 3.3** this pressure is trying to keep rams open. Assuming a mud weight of 9.2 ppg, a water depth of 5000 ft, and a 500 psi gas kick for a typical ram, this correction would be 160 psi.[3]

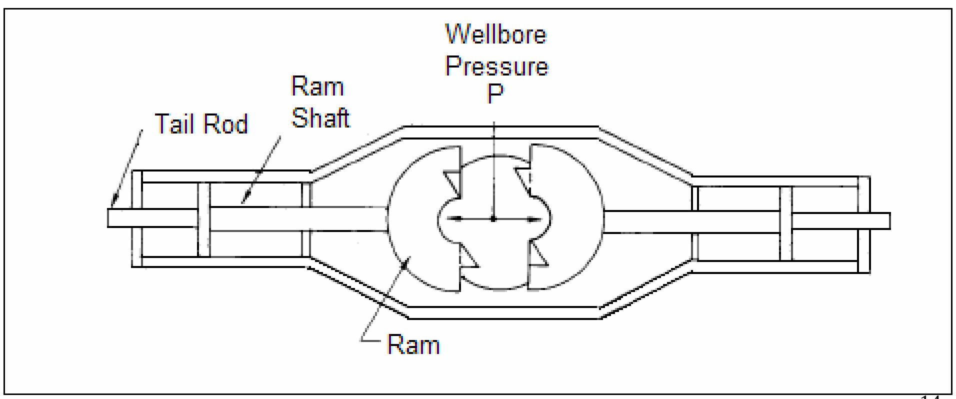


Fig. 3.3- Wellbore pressure resists against closing rams. (After Schlater[14])

The best applications of Eq. 3.11, which has been developed based on Boyles Low, are those of around atmospheric conditions. The greater the deviation from atmospheric conditions, the greater the discrepancy between Boyles Law and actual gas properties. So, Eq. 3.11 will not provide us with enough accuracy in deeper waters. The actual usable fluid in deeper waters is less than what Eq. 3.11 gives us. However, as drilling moves to ever-deeper waters, gas charged accumulator capacity calculations must become more sophisticated than the traditional ideal gas law method.

The following example helps us to figure out how an accumulator's volumetric efficiency will decrease in deepwater. Consider a nominal 15-gallon accumulator; actual volume is equal to 13.7 gallons. Let's first calculate this accumulator's usable fluid with atmospheric ambient pressure. Precharge pressure, maximum working pressure, and minimum working pressure respectively are 1,800, 5,000, and 2,000 psig. Substituting these values in Eq. 3.10, we would get 7.38 gallons for usable fluid:

$$\left(\frac{1{,}800+14.7}{2{,}000+14.7}-\frac{1{,}800+14.7}{5{,}000+14.7}\right)\times 13.7 = 7.38\,\text{gal.}$$

We repeat this calculation for a water depth of 3,500 ft. Substituting the proper values in Eq. 3.11, we will get 5.9 gallons usable fluid for the accumulator mentioned above:

$$\left(\frac{1{,}800+14.7+0.445\times 3{,}500}{2{,}000+14.7+0.445\times 3{,}500}-\frac{1{,}800+14.7+0.445\times 3{,}500}{5{,}000+14.7+0.445\times 3{,}500}\right)\times 13.7 = 5.9\,\text{gal.}$$

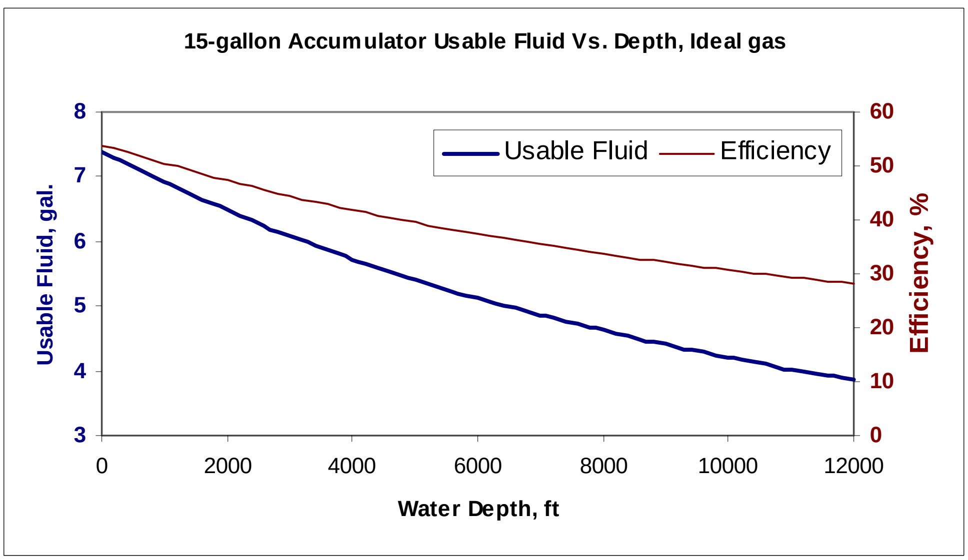


Fig. 3.4- Decrement in usable fluid versus water depth.

Clearly, when we are going into deeper water, we get the less volumetric efficiency for that accumulator. **Fig 3.4** makes this fact more clear. As we have declared before, volumetric efficiency of an accumulator is declared as the volume of usable fluid as a percentage of its actual fluid capacity.

3.6 Gas Compressibility Model

The problem will become even worse, if we install the accumulator in deeper water. We can not use Eq 3.11 any longer under these conditions. To get more accurate results, we have to switch to another method. We must include the gas compressibility factor into our calculations. This model gives us less usable fluid than the isothermal model but is more accurate.

At pressures of 5,000 psi and above, compressibility becomes a notable factor.[4] How much to reduce the pressure in order to determine a more accurate volume depends on the temperature and pressure.

To more accurately compute the usable volume in a theoretical accumulator, the ideal precharge pressure as well as the minimum and maximum pressures will have to be adjusted by the appropriate Z-factor. The effect of gas compressibility is incorporated into Eq. 3.12. The Z-factor can be determined by existing tables, graphs, or methods.

$$V_U = \left(\frac{P_n / Z_n}{P_{\min} / Z_{\min}} - \frac{P_n / Z_n}{P_{\max} / Z_{\max}} \right) \times V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.12)$$

Derivation of the equation above is presented in Appendix A. The change in temperature during charging and discharging time is neglected.

We use a typical deepwater drilling environment for the accumulator in the example; let's say the accumulator is installed at the depth of 7,000 ft. For this example pressures and corresponding compressibility factors are as follows;[4]

$$P_n = 0.445 \times 7{,}000 + 14.7 + 1{,}800 = 4{,}930 \text{ psia}$$
$$P_{\max} = 0.445 \times 7{,}000 + 14.7 + 5{,}000 = 8{,}130 \text{ psia}$$
$$P_{\min} = 0.445 \times 7{,}000 + 14.7 + 2{,}000 = 5{,}130 \text{ psia}$$
$$\mathrm{Z_n} = 1.19,\ \mathrm{Z_{max}} = 1.45,\ \mathrm{Z_{min}} = 1.2$$

We substituted these values in Eq. 3.12 and we got 3.2 gallons for the usable fluid volume of the accumulator whose usable fluid at the depth of 3,500 ft was 5.9 gallons. The value of 3.2 is not the exact usable fluid volume, but it is more accurate than the previous method (isothermal method) at that depth, and yet is proving that the usable fluid volume of gas charged accumulators dramatically decreases in deeper waters.

3.7 Adiabatic Non-ideal Gas Model

In deeper waters, higher pressure and full accumulator depletion produces rapid discharges, which suggest adiabatic gas expansion rather than the more common isothermal expansion process. Isothermal gas expansion assumes that changes in volume or pressure take place at constant temperature.[4] Functioning the BOP stack releases hydraulic fluid from the subsea accumulators, which allows the precharged gas to expand. When gas expands, its temperature decreases. So, in order for the gas temperature to be held constant and satisfy an isothermal expansion process, the gas would have to expand slowly. A slower expansion rate permits the gas to gain heat from the surrounding environment and, thus, maintain a constant gas temperature.

In contrast, an adiabatic process assumes no heat transfer. Practically any process can be made adiabatic if it occurs fast enough and higher gas precharge pressures tend to produce higher discharge rates.[4] With BOP functions occurring in the 20 second range, there is minimal time for significant heat transfer.[4] Therefore, it is logical to assume that the relatively quick discharge of gas in subsea accumulators at higher pressures tend toward an adiabatic process.

We calculate usable fluid for the accumulator mentioned above to help us figure out the significant difference in results between adiabatic and isothermal model. Considering the same parameters but assuming an adiabatic expansion process, the amount of usable fluid in a 15-gallon accumulator is determined to be only 2.2 gallons. Eq. 3.13 is used to calculate the usable fluid in this model.[6] In this method, gas is considered as a non-ideal gas, and so we have adjusted pressures with compressibility factor (*P/Z*) before using them in Eq. 3.13.[4] Refer to Appendix A to see the derivation of Eq. 3.13.

$$V_U = \left(\left(\frac{P_n}{P_{\min}} \right)^{0.71} - \left(\frac{P_n}{P_{\max}} \right)^{0.71} \right) \times V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.13)$$

3.8 Cameron Model

Cameron investigators believe that typical discharge times allow enough heat to flow between the gas and the accumulator to cause serious deviations from both isothermal and adiabatic models. So, they used their own model to calculate usable fluid of accumulators located at ever-deeper waters. The model includes both gas compressibility and effects of heat transfer between the gas and accumulator bottle.[1] They claim that the results are particularly helpful in design of any accumulator system operating at pressures of 5,000 psi and higher. As can be seen from **Fig. 3.5**, there is not a significant difference between the adiabatic expansion calculation and results of Cameron's computer program at high pressures.

The best, and thus recommended, method of calculating gas behavior is to assume constant entropy, or adiabatic expansion. Entropy is the measure of order in a system; this correction assumes that in changing temperature and pressure while the gas expands, the order remains constant. This is a textbook thermodynamic assumption that fits the physical situation well. From the starting entropy (at a given temperature and pressure), the gas is expanded adiabatically to the target pressure, P_{min}. The density of gas at this condition is then read. For precharge pressure, one minus the ratio of beginning to ending densities at the two conditions results in the usable fluid fraction.[3]

However, as can be seen from **Fig. 3.5**, the usable fluid fraction dramatically decreases when going to the deeper water. To compensate for this, a large number of accumulators should be mounted in the hydraulic unit at the seafloor. Fig. 3.5 shows that usable volume fraction at the depth of 10,000 ft may be 0.12. It means that an accumulator's usable fluid at that depth should be 0.12 of its actual fluid capacity.

Usable fluid fraction of accumulators in use is used to calculate the number of accumulators required at the depth of interest. To calculate the number of accumulators, we must divide the volume of hydraulic fluid required to manipulate the stack and meet the regulatory agency requirement by the calculated usable fluid fraction. As an example assume that we have calculated this volume of fluid to be 265.5 gallon. Since the usable fluid fraction decreases as water depth increases, clearly the number of accumulators increases (**Fig. 3.6**). As Fig. 3.6 demonstrates, based on adiabatic method calculation which is the most accurate method of calculation of deepwater accumulator usable fluid, we need more than 150 accumulators at water depth of 10,000 ft.

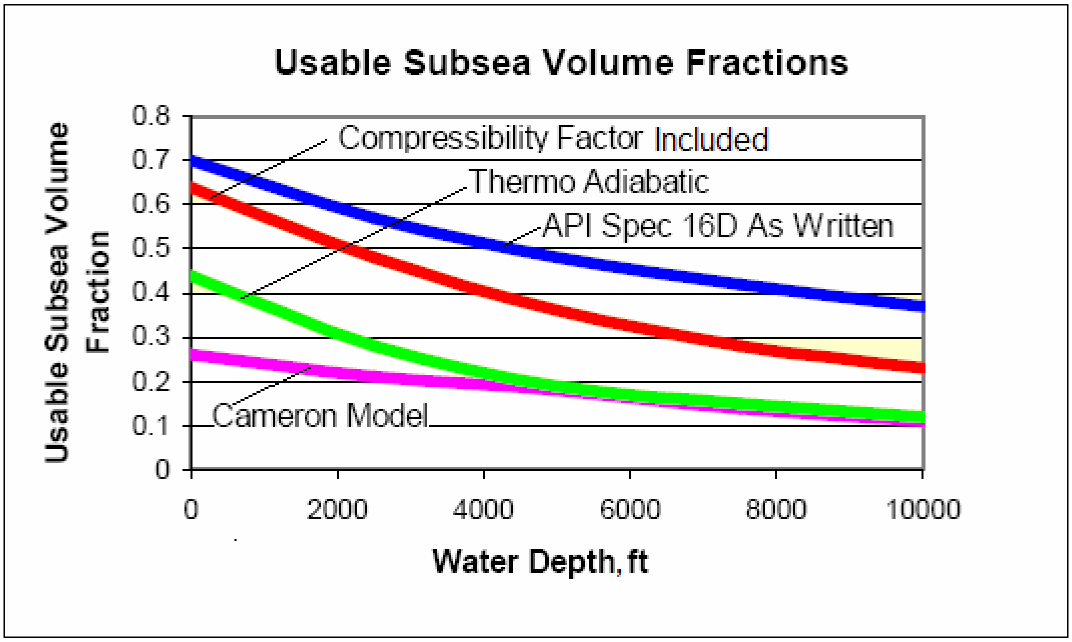


Fig. 3.5- Comparison of different models of usable fluid calculation.[3]

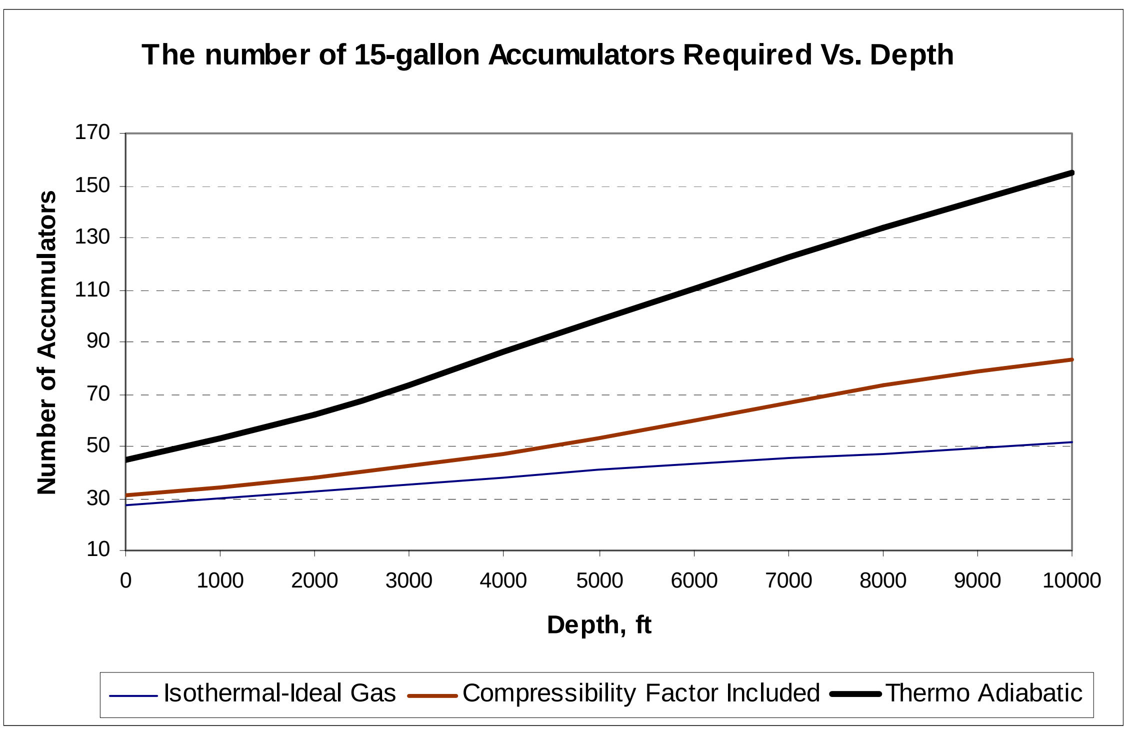


Fig. 3.6- The number of accumulators increases in deeper waters.

CHAPTER IV

POSSIBLE REPLACEMENTS FOR CONVENTIONAL ACCUMULATORS

4.1 Spring Loaded Accumulators

The idea of using a spring instead of gas in accumulators comes from the fact that a deflected spring can store energy in it and this energy can be recovered at the time we need. And unlike the gas, the energy stored in a spring is not a function of its ambient pressure. A spring stores the same amount of energy at any water depth as it would do on the surface.

4.1.1 Energy Storage

A mechanical spring is defined as an elastic body which has the primary function to deflect under load, and to return to its original shape when the load is removed.

The external work imported to the spring by the product of two factors, load and deflection, is stored by the spring as potential energy and may subsequently be recovered (except for frictional losses) as kinetic energy. For coil spring applications, the deflection, *f*, can be considered as proportional to the load, *P*, provided the elastic limit of material is not exceeded. If the spring is considered as having a constant rate *R* with load expressed in terms of lbf and deflection expressed in terms of inch, then the energy stored by the spring at deflection *f* or load *P* can be expressed in terms of lbf-in as follows:

$$U = \frac{1}{2}Pf = \frac{1}{2}Rf^2, \quad \text{(4.1)}$$

where *R*, the rate of spring, is expressed in terms of lbf/in. If you need to get energy in terms of Joules, you just divide *U* by conversion factor of 8.8512 lbf-in/Joules. The preceding formula is based on the fact that deflection starts from the free length or free position of the helical spring. If the spring is given some initial deflection, f_1, which develops load, P_1, from this point, the increase of the load to its maximum value P_2 will cause the spring to be deflected through a distance or stroke, f_2 - f_1, to the maximum deflection f_2. The subsequent decrease in load to P_1 will return the spring to its assembled length. In such conditions, the energy to be absorbed by the spring during this cycle is:

$$U = \frac{P_1 + P_2}{2}\left(f_2 - f_1\right) = \frac{1}{2}R\left(f_2^{\,2} - f_1^{\,2}\right), \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4.2)$$

Generally, we can define resilience for a body. Resilience is the potential energy stored up in a deformed body. The amount of resilience is equal to the worked required to deform the body from zero stress to stress *S*, when *S* does not exceed the elastic limit.[15] When we compress a helical spring, indeed we are deforming the wire which the spring is made of, that’s why a deflected spring can store energy in it.

The general problem for a designer is to know the overall energy capacity required of the spring and to relate this to maximum stress in the spring. It has been established in fundamental studies that the energy capacity of a spring for a given maximum stress increases in direct proportion to the volume of the active spring material. For a cylindrical helical spring of circular cross section (**Fig. 4.1**), which is subject to torsional stress, the energy capacity per unit volume of active material is as follows:[15]

$$U_p = \frac{U}{V} = \frac{S^2}{4G}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4.3)$$

where G is the modulus of elasticity of the material in shear stress (shear modulus of elasticity). Since the helical spring is subject to shear stress, we should use the shear modulus of elasticity; G. In Eq. 4.3, U_p is used as the basic index for the efficient utilization of spring material.[16] The formula indicates that the specific energy capacity increases with the square of the maximum shear stress which the material used in a helical spring can withstand without permanent set.

Specific energy capacity is also inversely proportional to the modulus of elasticity of the material.

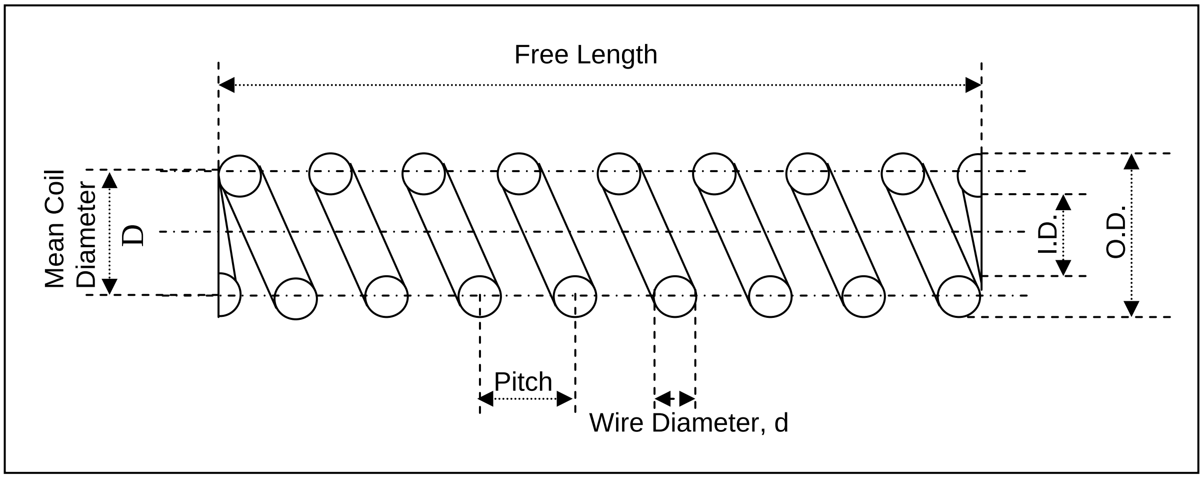


Fig. 4.1- Cylindrical helical spring of circular cross section.

4.1.2 Structure and Operation

A spring loaded accumulator may look like what you can see in **Fig. 4.2**. When the power fluid is pumped into this kind of accumulator, the spring inside the accumulator is deflected by the piston which separates the fluid side of the accumulator from its spring side.

The piston must be completely sealed against the fluid leakage to the spring side of the accumulator. This problem has already been solved in the industry. Parker, one of the manufacturers of accumulator in the U.S., uses a V-O-ring to seal the piston inside the piston type of gas charged accumulators. In this kind of accumulator, the piston seal consists of a unique, patented five bladed V-O-ring with back-up washers.[5] This design eliminates seal roll-over and ensures total separation of the fluid side and spring side of the accumulator. The V-O-Ring also holds up full pressure throughout long idle periods between cycles, providing dependable, full pressure storage of hydraulic energy. It ensures safe, reliable absorption of pressure peaks.[5]

Parker also uses PTFE glide rings to eliminate metal-to-metal contact between the cylinder and the piston, reducing wear and extending service life (**Fig. 4.3**).

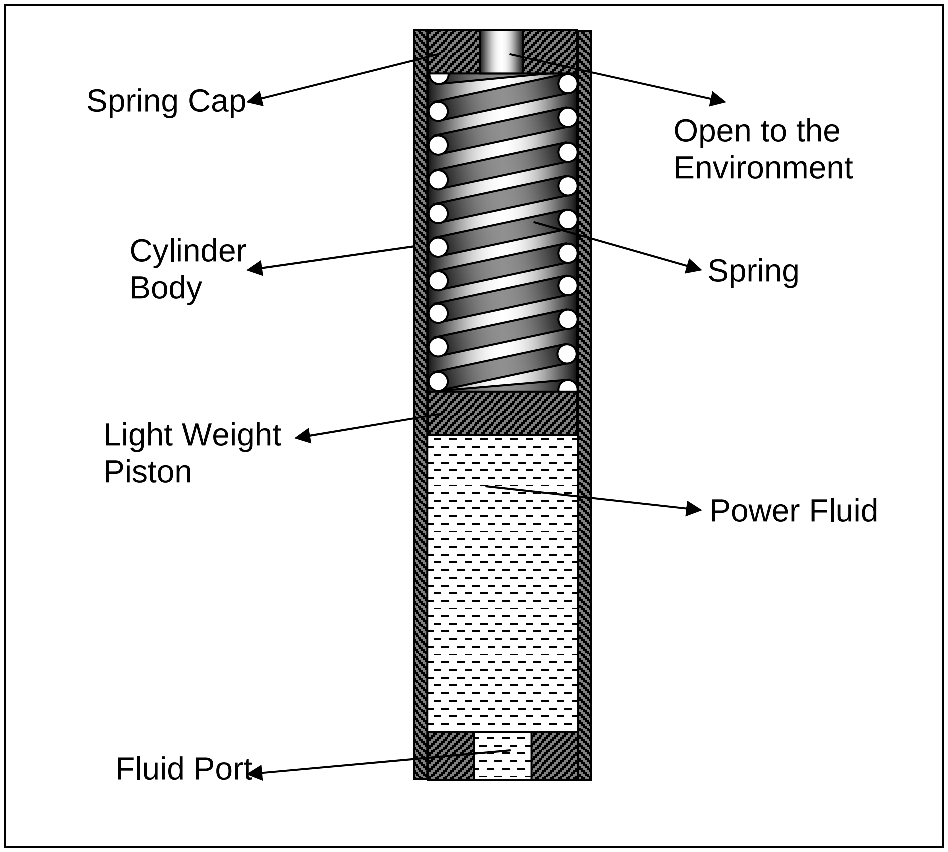


Fig. 4.2- Subsea spring-loaded accumulator.

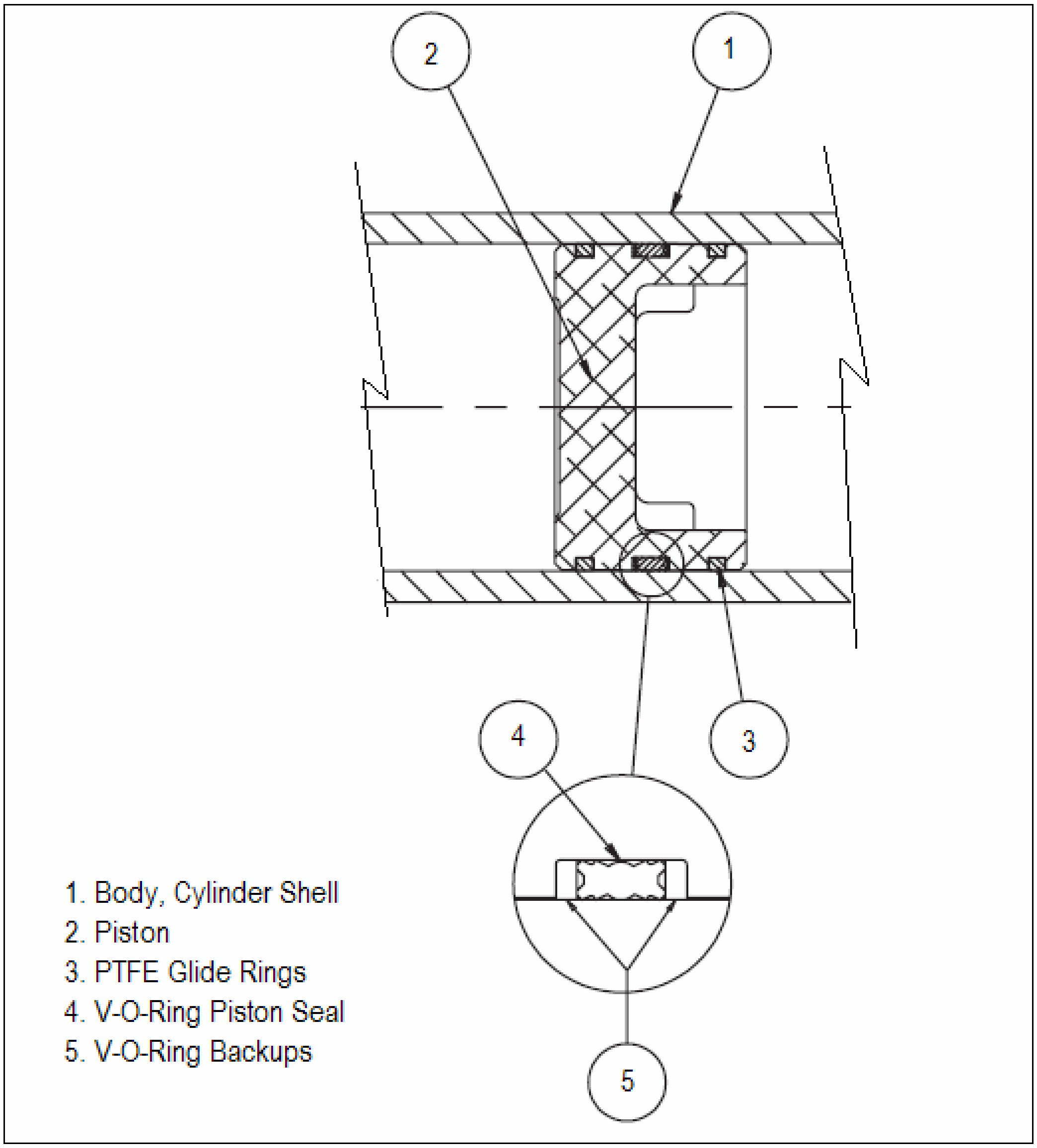


Fig. 4.3- Sealing system around a piston.

Operation. Before starting to pump fluid into the fluid side of this kind of accumulator, the spring is in its free condition (it is not compressed yet). It means that we do not have any precharge pressure in spring charged accumulators, in contrast with gas charged accumulators. In a gas charged accumulator we need to precharge the accumulator to prevent piston in piston type or bladder in bladder type from damage when they are charged to the maximum working pressure. Without pressured gas in the gas side of gas charged accumulators it is impossible to get the predetermined maximum working pressure inside the accumulator.

Like conventional accumulators, spring accumulators should keep 200 psi or more above the precharge pressure in the system after it has been discharged, as API 16D mandates. It means that the spring should not extend to its free original length. **Fig. 4.4** shows operation of spring loaded accumulators.

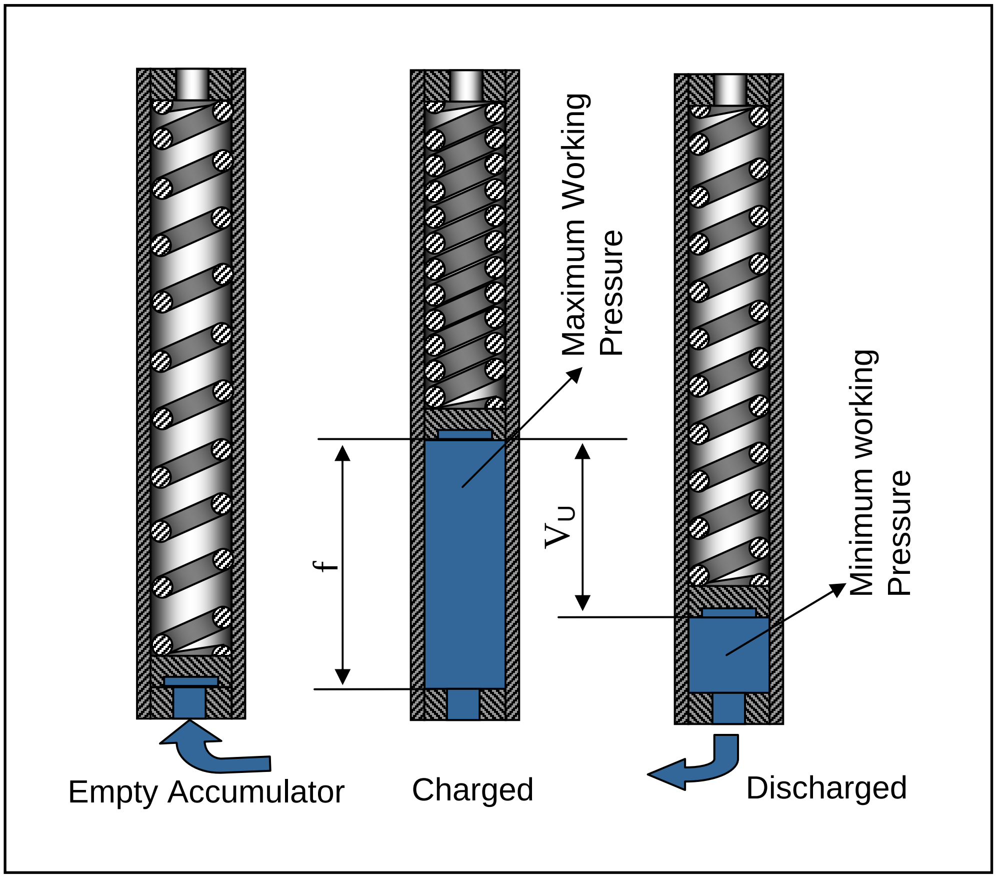


Fig. 4.4- Operation of spring charged accumulators.

4.1.3 Spring Design

Our primary objective in a spring design is generally to obtain a spring which is able to be loaded high enough to provide maximum working pressure. This spring should fit into a reasonable sized cylindrical body of spring loaded accumulator. This accumulator must be able to store the highest possible volume of power fluid, so that we can reduce the number of accumulators as much as possible. Since the spring is to be exposed to the corrosive sea water, we have to consider this factor in spring material selection.

The first step in the design is to determine the load and deflection for a given maximum working pressure and the quantity of usable fluid required.

The following formulas are used to design helical compression springs of round wire.[17,15] See Appendix B for derivations.

$$S_v = \frac{8\,P\,D}{\pi\,d^3}, \quad \text{(4.4)}$$

$$S_v{}' = \frac{8\,P\,D}{\pi\,d^3}k, \quad \text{(4.5)}$$

$$k = \frac{4c-1}{4c-4} + \frac{0.615}{c}, \quad \text{(4.6)}$$

$$c = \frac{D}{d}, \quad \text{(4.7)}$$

$$f = \frac{8\,n\,D^3 P}{d^4 G}, \quad \text{(4.8)}$$

where P is load on spring, D is mean diameter of spring (Fig. 4.1), d = wire diameter, f = deflection, S_v = maximum allowable shear stress in wire, S_v' = corrected shear stress (see Appendix B), k = Curvature correction factor (**Fig. 4.5**), and n is the number of active coils in the spring.

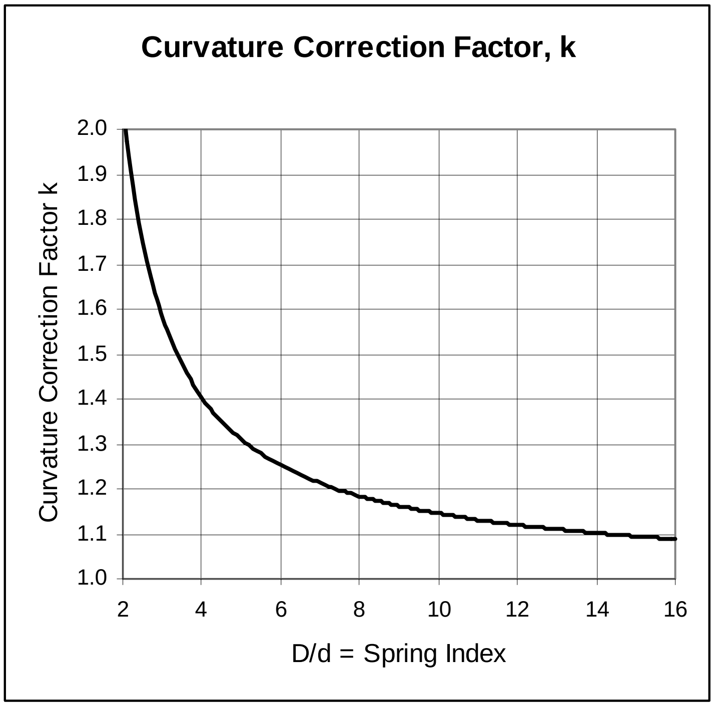


Fig. 4.5- Curvature correction factor for helical round wire compression springs.

Load on the spring, *P* (F_s). Fig. 4.6 shows a free body diagram of a piston inside a subsea spring charged accumulator.

As can be seen from Fig. 4.6, equilibrium equation of forces is as follows:

$$F_s + P_{hydrostatic} A = \left(P_{\max} + P_{hydrostatic}\right) A, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4.9)$$

We solve this equation for F_s:

$$F_s = P_{\max} A, \quad \text{(4.10)}$$

F_s is the load on the spring that is directly proportional to the maximum working pressure and square of piston diameter:

$$F_s = \frac{\pi}{4} P_{\max} D_p^{\ 2}, \quad \text{(4.11)}$$

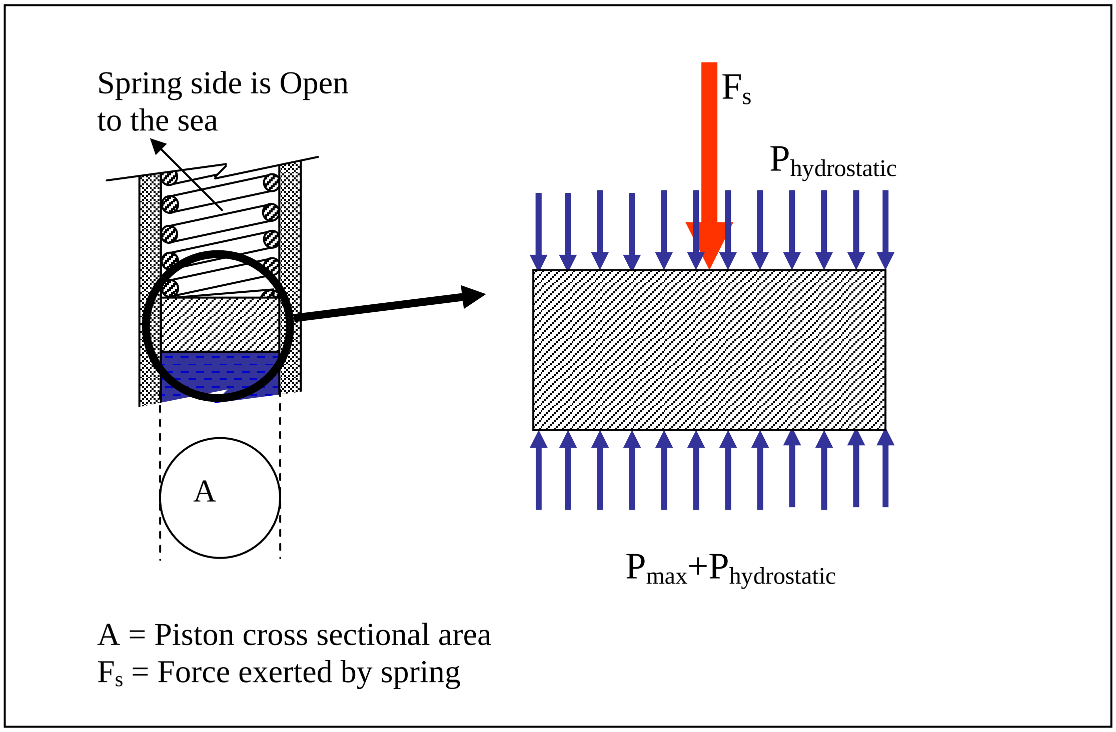


Fig. 4.6- Free diagram of piston in a subsea spring charged accumulator.

Deflection of spring, *f*. Deflection required in the spring can be determined based on the volume of stored power fluid needed in one accumulator unit. With a constant piston cross sectional area, the deflection of the spring is directly proportional to the usable fluid:

$$V_p = A \times f, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4.12)$$

where A is the cross sectional area of the piston and V_p is the volume of fluid pumped into the accumulator to get the maximum working pressure at deflection f.

It can be concluded from Eq. 4.12 and Eq. 4.10 that A, V_p, f, and the load on the spring P, are dependent on each other and none of them can be determined separately. It means that any changes in one of them would change the other parameters. Any changes in the diameter of the piston changes the load on the spring and f.

Material. Referring to Eq. 4.10 and substituting some common values for P_{max} and A, we can find that the spring we are going to design needs to withstand a really huge load; may be a load higher than the load which helical springs in available heavy equipment currently bear. Our primary objective of material selection is to choose a spring material which is used in manufacture of heavy duty springs.

The alloy spring steels have an important place in the field of spring materials, particularly for conditions involving high stress and where shock or impact loadings occur. Alloy spring steels can also withstand both high and low temperatures. All these alloys are used in springs for equipment in wire sizes frequently under ¼ in. diameter and up to 1/2 in. Since we need to design a spring to meet a high load, we should select a material whose larger bar sizes are available too. The alloy steels are generally recommended for the larger sizes and higher stresses.[17] Annealed bars of alloy springs are available from 3/8 to 2 in. or larger.[18] Hot-rolled Nickel-Chromium-Molybdenum alloy steel bars, ASTM A 331, are available up to 2.5 in. in diameter.[18] For more information about spring material selection go to Appendix C.
Helical compression springs having bar diameters larger than about 5/8 in. are commonly coiled hot and then heat-treated, since it is not practical to wind such springs cold.[17] So, the tentative spring we intend to design would be kind of hot-wound

compression spring. Currently, hot-wound springs are used in automotive, railroad, armament, and heavy equipment.

Shear modulus of elasticity, *G*. In figuring deflections and stresses, a modulus of rigidity of 10.5×10^6 psi is frequently used by spring manufacturers for alloy steel bars in hot coiled springs.[17] Carlson suggests 10.75×10^6 for G in his book.[18]

Some processes may reduce the modulus G. Overstraining of the spring such as occurs in presetting tends to reduce this modulus. However, it would be accurate enough to use these values in order for our primary analysis of deflection, and load of spring, and consequent shear stress in spring wire.

Working stress, S_v. Carlson in his book,[18] presented the elastic limit of materials as a percentage of their ultimate tensile strength. For alloy steel bars where they are in torsion, the maximum shear stress without permanent set should be 60 to 70 percent of ultimate tensile strength. Ultimate tensile strength for alloy steel bars varies from 180,000 to 200,000 psi.

There are some graphs in some topic related books and manuals, which suggest working stresses with respect to bar diameters.[15,17] Maximum allowable working stress varies with bar diameter; allowable working stresses are higher for smaller bar diameters.[15,17] However, we use the values suggested by Carlson and Wahl for our primary analysis.

Now, let's get into design of a helical compression spring for our tentative accumulator. **Table 4.1** is based on the formulas given above in this chapter. d is diameter of wire, D = Pitch diameter (center to center of wire), P = safe working load for the maximum allowable stress, and f is deflection of one coil for the safe working load P.

Table 4.1- Safe working load *P* and deflection *f* for hot-wound helical compression spring.

Maximum Allowable Shear Stress = 108,000 psi, G = 10,750,000 psi														
d, in.	D, in.	3	4	5	6	7	8	9	10	11	12	13	14	15
1.0	P	8,948	7,553	6,473	5,644	4,995	4,477	4,055	3,705	3,409	3,157	2,940	2,750	2,583
	f	0.184	0.368	0.616	0.929	1.305	1.747	2.252	2.823	3.457	4.157	4.921	5.749	6.642
1.5	P		21,292	19,010	16,995	15,302	13,887	12,698	11,689	10,823	10,074	9,421	8,845	8,335
	f		0.205	0.358	0.552	0.790	1.070	1.393	1.759	2.168	2.620	3.115	3.653	4.234
2.0	P			38,865	35,790	32,845	30,213	27,906	25,890	24,126	22,574	21,202	19,982	18,891
	f			0.231	0.368	0.536	0.737	0.969	1.233	1.529	1.858	2.218	2.611	3.036
2.5	P				61,634	57,855	54,031	50,457	47,208	44,285	41,662	39,307	37,187	35,272
	f				0.260	0.387	0.540	0.717	0.921	1.150	1.404	1.684	1.990	2.322

High pressure in spring charged accumulator can be obtained by exerting a high force on the piston which is separator of spring and liquid. Table 4.1 shows that, the maximum possible force obtained from a helical compression spring would be 61,634 lbf. It is not exactly the maximum force a deflected compression spring may provide, but it is accurate enough not to mislead us. Even a safe load of 100,000 lbf would not solve our problem.

The table above shows that the pitch diameter of this spring is 6 in. So outside diameter of this spring would be 8.5 in. (6 in. + 2.5 in. = 8.5 in.). Consider 8.5 in. (outside diameter of spring) as diameter of piston in our accumulator. So, the pressure which this spring can provide us would be:

$$P_{\max} = \frac{P}{A} = \frac{61{,}634}{56.75} = 1086\,\text{psi, where } A = \frac{\pi}{4}8.5^2 = 56.75\,\text{in}^2.$$

This pressure is not good for hydraulic BOP control purposes. Common maximum working pressure for subsea BOP control systems would be 5,000 psi. Of course we can put the spring in an order with other parts to get that much pressure. But it is not based on sound engineering principles and practices. The force of 61,634 lbf should be exerted on an area of 12.327 in^2 to provide a pressure of 5,000 psi (**Fig. 4.7**). In this case, to store just 5 gallons of hydraulic fluid the stroke of the piston inside the cylinder should be 94 in. It means that the maximum deflection of spring should be 94 in. Since the deflection of one coil of spring mentioned above is 0.260 in. (Table 4.1), we need 361 (94 / 0.260) coils of spring. Regarding that the bar diameter is 2.5 in., solid length of such spring would be 902 in. (2.5 × 361). Since the length of this spring is 996 in. (902 + 94), the height of this accumulator would be 1090 in. (996 + 94) that is really too long and is not feasible.

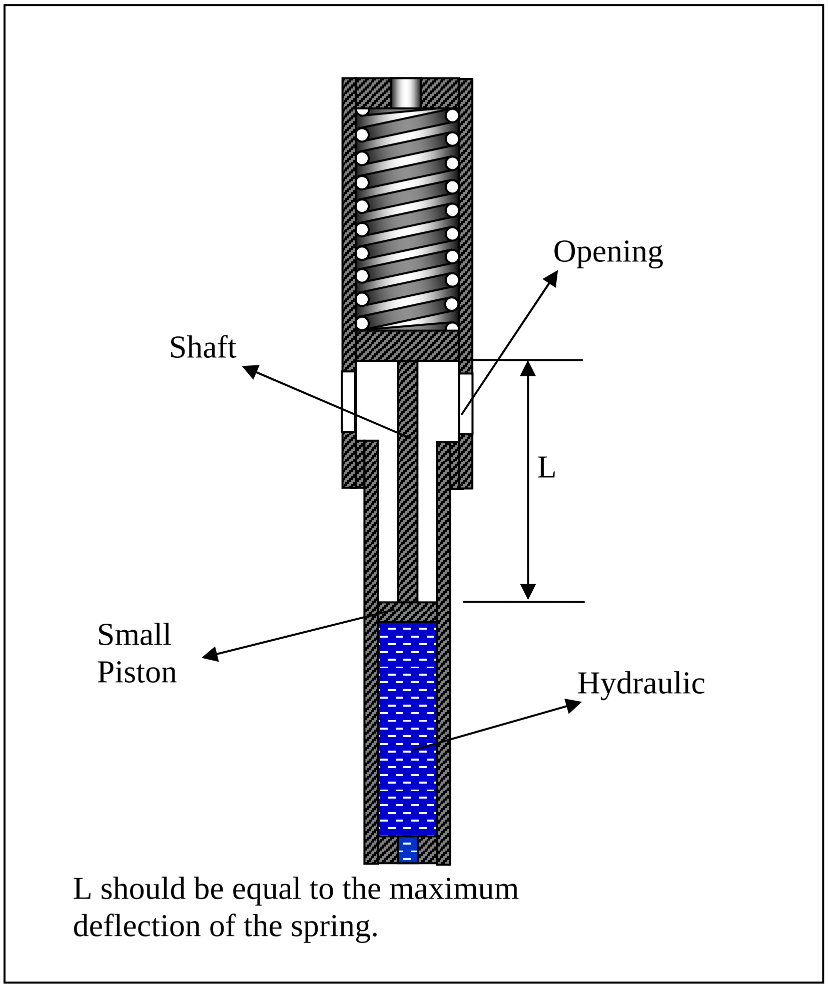


Fig. 4.7- Smaller piston to get higher pressure.

We added 94 to 996, because it is the length of the shaft that connects the larger piston to the small piston (L = 94 in.).

Based on Eq. 4.11 the load which we need the spring to exert on the piston to provide us with the pressure of 5,000 psi is 283,725 lbf which is not really a load which a compression spring can bear. Indeed, there is no helical compression spring that can tolerate this load.

Application of spring loaded accumulators in subsea BOP control systems may be more disappointing, if we use energy storage of springs to analyze feasibility of spring accumulator. Maximum energy storable in cylindrical helical spring of circular cross section can be calculated by Eq. 4.13.[15]

$$U = \frac{S_v^{\,2} V}{4Gk^2}, \quad \text{...} \quad (4.13)$$

As we can see in Appendix D, 7.36×10^6 Joules of energy must be released from 24 subsea gas charged accumulators to close one unit of annular preventer at the depth of 7,000 ft. Now, the volume of material required in the shape of springs to store that much energy can be calculated by Eq. 4.13. If we put the mechanical properties of the spring mentioned above in Eq. 4.13 and solve it for V, we will get 1,383,322 in^3 for it. It means that we need that volume of coil spring material to coil to store 7.36×10^6 Joules energy. Knowing that the density of material is 0.285 lb/in^3, we can find that more than 178 tons of spring material is required to store that much energy.

The volume of material will be still high if we use another type of cylindrical helical spring: rectangular cross section spring. In this kind of spring the cross section of wire looks like a rectangle. Energy formula (Eq. 4.14) is used to know how much volume of spring material we need.

$$U = \frac{4S_v{}^2V}{45G}\left(1 + \frac{1}{k^2}\right), \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4.14)$$

If we use the values of 108,000 psi, 10.75×10^6 psi, and 1.792 respectively for S_v', G, and k, in Eq. 4.14, we will find that more than 66 tones of this kind of spring is required to store energy required to do just one function on the BOP stack.

This analysis even rejects the idea of nested spring to provide high force and subsequently high pressure.

4.2 Weighted Accumulator

When we are lifting an object, we are doing work on it. This work done on the lifted body is stored as potential energy in it and can be recovered as kinetic energy. This energy can be calculated by Eq. 4.15.

$$W = wh = mgh, \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (4.15)$$

where w is the weight of object, m is mass, g is gravity acceleration, and h is vertical displacement of object.

In this thesis, in order to perform a feasibility analysis of weight charged accumulators, we compare it to conventional accumulators by means of energy. Energy required to open one unit of annular preventer is calculated in Appendix D to be 7.36×10^6 Joules.

Weighted accumulators should store that much energy to do the same function on the BOP stack. It means that $mgh = 7.36 \times 10^6$ Joules. If we construct an equipment to displace the mass vertically as high as $h = 15$ ft, we need to provide the equipment with a body of 361,770 lbm to be lifted to 15 ft above its original location to get that much potential energy. If we use lead with the density of 0.410 lb/in^3 (0.375 lb/in^3 in water) in our tentative accumulator, we will need to mount 560 ft^3 of lead at the sea floor just for doing one function on the BOP; closing annular preventer.

There is another problem with this kind of accumulator; this mass should exert its weight as vertical force on a piston with a cross sectional area small enough to provide us the pressure required in subsea BOP control system, for example 5,000 psi. A piston with a cross sectional area of 50 in^2 (8 in. diameter) should be loaded by 250,000 lbf (249,912 lbm), which is 386 ft^3 of lead. Maybe a machine like what you see in **Fig. 4.8** can do that for us, which is not a good replacement for gas charged accumulators. May be these weighted accumulators had been the first shape of accumulators which were replaced by gas charged accumulators in early days when accumulators were needed.

Fig. 4.8- Heavy weight required in weighted accumulator.

CHAPTER V

CONCLUSION

Since the volumetric efficiency of gas charged accumulator drops in deep waters, we need a large number of this kind accumulator to do functions on the BOP stack at the sea floor. In this thesis we evaluated the feasibility of replacing these accumulators by spring or weight loaded accumulators to reduce the number of accumulators required. The following conclusions can be derived from this study.

1. It is possible to build spring loaded accumulator, but this kind of accumulator is not practical for BOP control purposes where a large amount of energy and high fluid pressure is required.
2. Replacement of gas charged accumulators by weighted accumulators is not feasible; tons of the heaviest and yet cheapest metal is required to be lifted very high to store the energy needed to performe the functions on the BOP stack.

CHAPTER VI

RECOMMENDATIONS

6.1 Electromagnetic Rams

Accumulators along with the other parts of the BOP control system are designed to get enough volume of pressurized hydraulic fluid in one side of a piston, which is connected to a shaft, to run the shaft and consequently the ram inside the BOP stack. What if we eliminate accumulators and give up the hydraulic power and use electromagnetic force to run the ram inside the BOP stack? We attempt to generate mechanical forces such that the moving part, which is the shaft of the ram, can perform work.

6.1.1 Rail-Ram

One of these forces is the force on a current carrying conductor in a magnetic field. Magnetic field, which is a vector field, can exert a force on a current carrying conductor in it. Laplace's law is used to calculate this force;

$$d\boldsymbol{F} = d\boldsymbol{l} \times \boldsymbol{B}\, I, \quad \text{(6.1)}$$

Where the vector $d\boldsymbol{F}$ is force, $d\boldsymbol{l}$ is a vector with the direction of the current and magnitude of dl (the length of the differential segment of the conductor), $\boldsymbol{B}$ the magnetic flux density, and I the current. From the vector notation, we note that the force is perpendicular to the plane formed by the vectors $d\boldsymbol{l}$ and $\boldsymbol{B}$ and its magnitude is

$$dF = BIdl\sin\theta, \quad \text{(6.2)}$$

where the θ is the angle between the vectors $d\boldsymbol{l}$ and $\boldsymbol{B}$ as in **Fig. 6.1**.

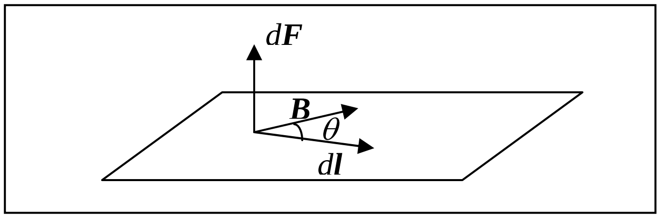


Fig. 6.1- The direction of the force ***dF***.

We are trying to use this force to actuate rams (**Fig. 6.2**).

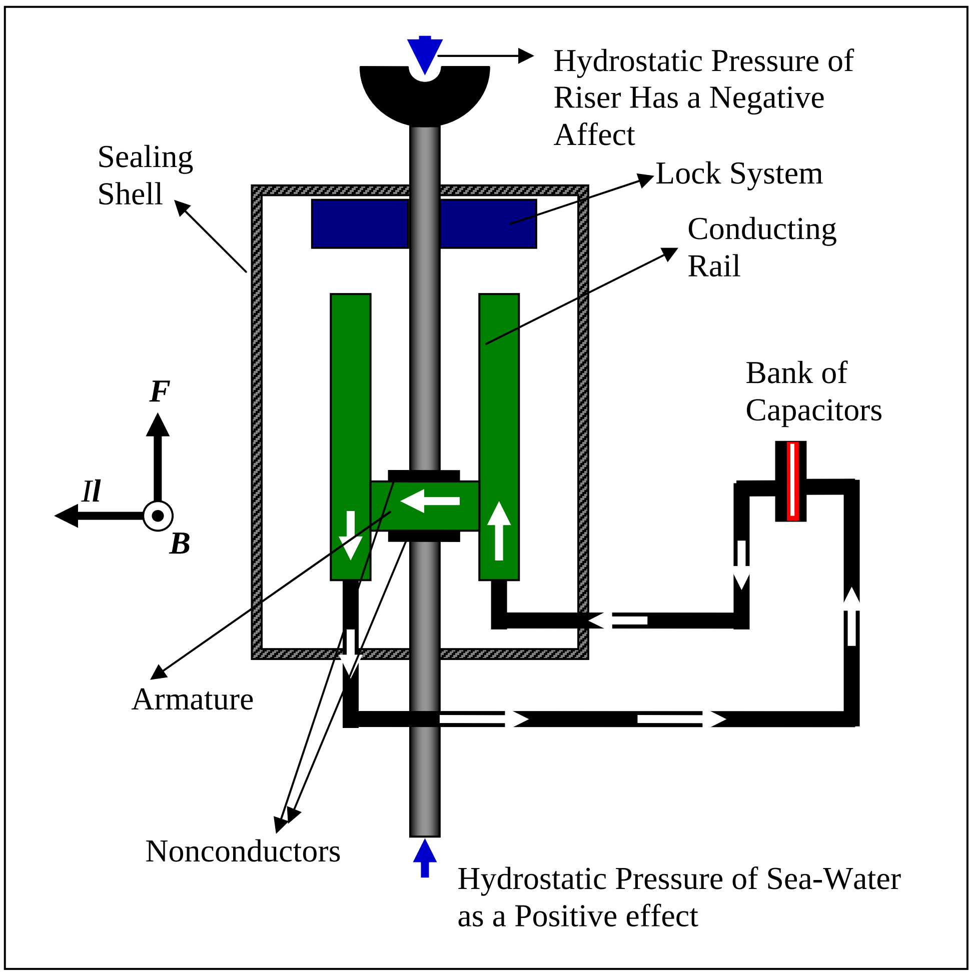


Fig. 6.2- A schematic of rail-ram.

We intend to use this force just to close the ram. The differential pressure between inside the riser and sea-water would be high enough to push the ram back to its original position, or a reverse current can open the ram, whichever is better.

The rails, in this device not only provide guidance for the armature, but also they serve to generate the magnetic field around the armature. To reduce the current, we may use a pair of windings to generate the magnetic field around the armature (**Fig. 6.3**).

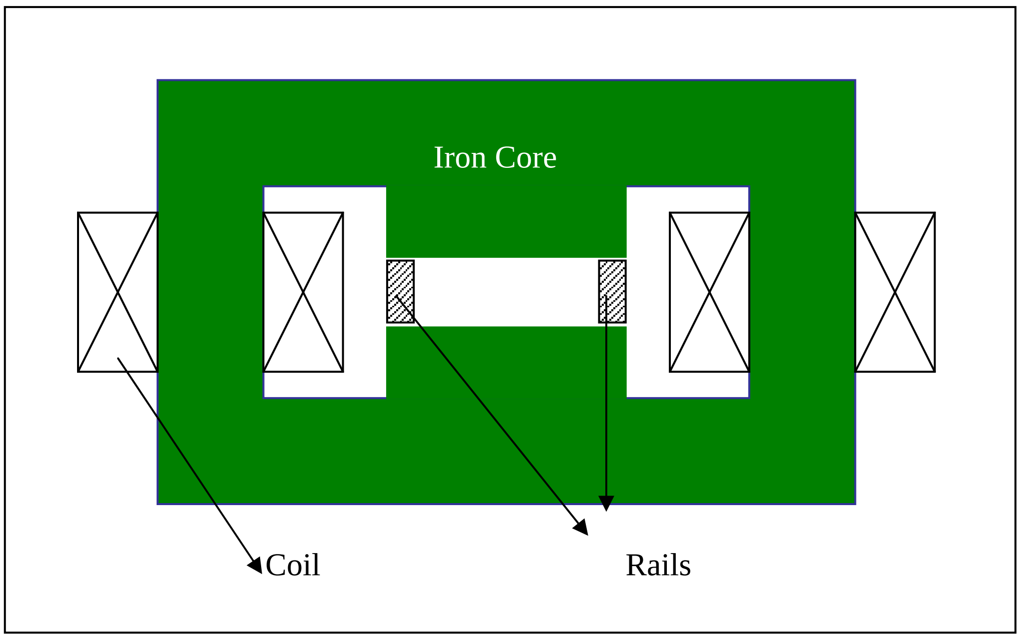


Fig. 6.3- Electromagnetic in order to generate a magnetic field around the rails.

In this system, when the ram is closed, the switch of the electric circuit should be open because of the high amount of heat that this system may generate. So we have to look for a way to lock the ram to keep it closed (**Fig. 6.4**). The locks in this system also can be actuated electrically.

The US navy is trying to use this technology to launch projectiles as a replacement for conventional guns on the ship. They developed and exercised a rail-gun parametric model and concluded that the rail-gun appears feasible.[20] In this gun, they reached a peak acceleration of 45,000 Gees. In the rail-gun, the total displacement of the armature is 39.37 ft, but in the rail-ram we are looking for a displacement of 9 in.

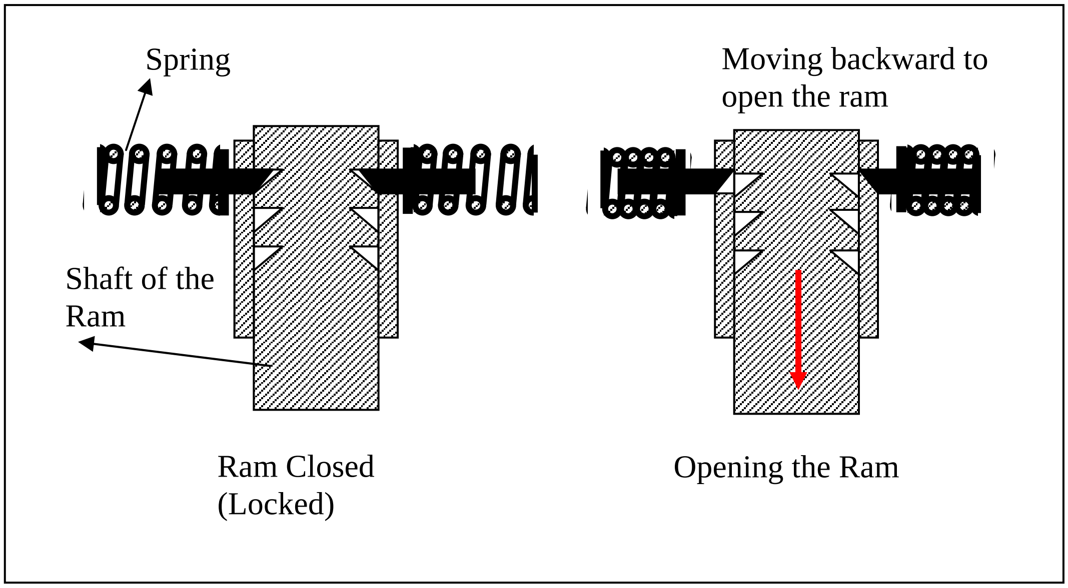


Fig. 6.4- Locking system to keep the ram closed.

The interaction between electrical and mechanical quantities is complex and must be approached carefully. There is no single, general method of treatment which can, in a simple manner, define this conversion of energy. It has been shown that for each type of device, there is a preferred method of treatment.[21]

The finite element method may be used to design such electromagnetic devices. The development and use of the finite element method to solve this problem requires the joint use of three distinct sciences:

- Electromagnetic science; to establish the governing equations related to this problem.
- Numerical methods; needed to apply the finite element concepts and solve the system.

- Computer sciences; to build and maintain efficient software tools needed for implementation.

This is currently just an idea and there are lots of question to be answered.

6.1.2 Magnetic Repulsion Rams

We got the idea of using magnetic repulsion force from the suspension system in magnetic levitation trains (Maglev). **Fig. 6.5** shows an electromagnet used to suspend and guide a current Maglev. Around the iron core, there are two windings with 187 turns; the two windings are in a series connection. With nominal current of 40 Amps, the electromagnet will keep the nominal air-gap to reach the nominal levitation force of about 529 lbf.[22]

The levitation force in such an electromagnet depends directly on the square of the current in the windings and the square of the number of turns of the windings.

We intend to use this repulsion force in another way; we are thinking of arranging some electromagnets in a line with no air-gaps between them when the ram is open, and the maximum air-gap when the ram is closed (**Fig 6.6**).

As soon as the switch closes the circuit and the current flows in the windings, the electromagnetic units repel each other to get to their maximum designed air-gaps. It is necessary to mention that the direction of the currents in the winding of each unit should be in the opposite direction of the neighboring one, so that similar electromagnetic poles generated at the end of each unit are next to each other. We know that similar magnetic poles repel each other.

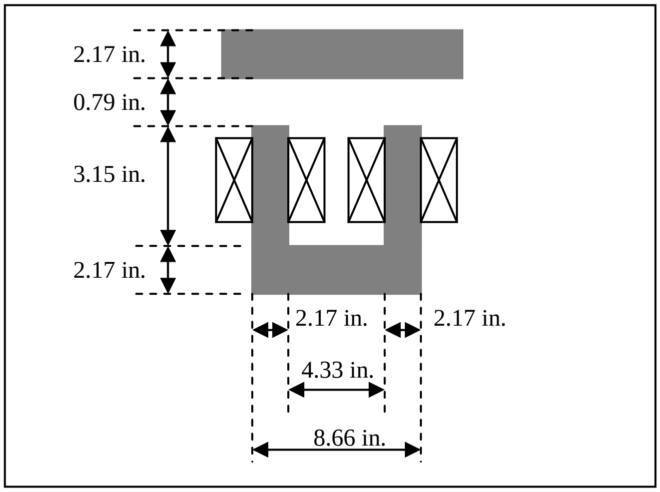


Fig. 6.5- Levitation and guidance magnet details.

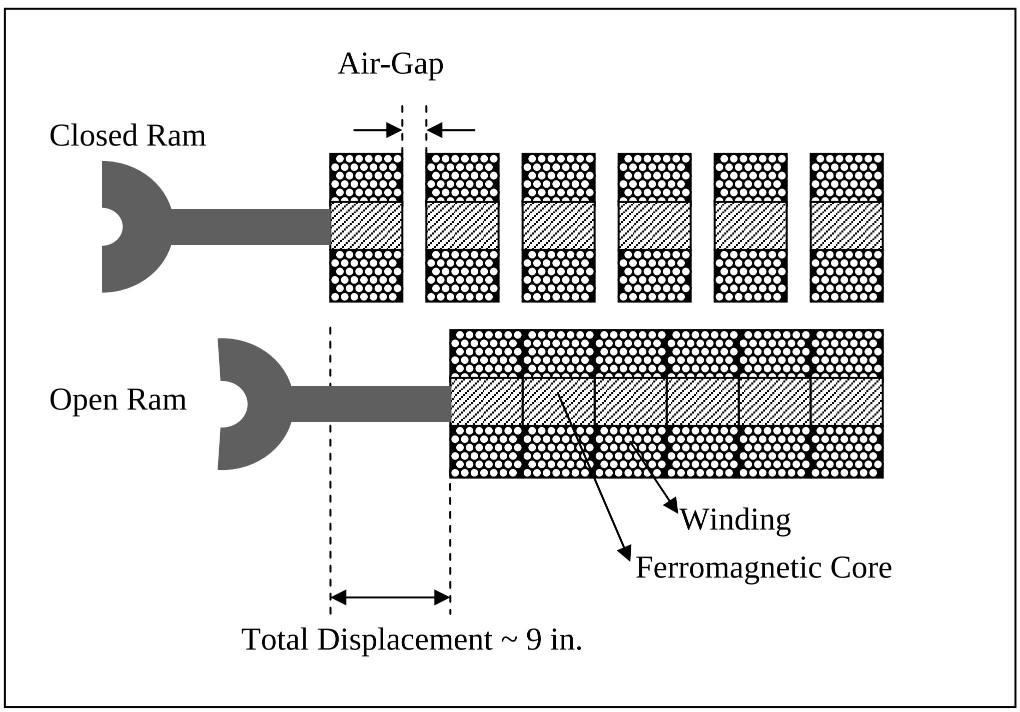


Fig. 6.6- A sketch of a magnetic repulsion ram.

6.2 Low Pressure Tank

In conventional BOP control systems, the hydraulic fluid is supposed to exhaust to the seawater each time we perform a function on the BOP stack. That is the reason why we have to include the hydrostatic pressure of water in the usable fluid calculation. As a matter of a recommendation, a low pressure tank located on the sea-floor can dismiss the negative effect of high hydrostatic pressure of seawater. If we connect the directional control valves, which are installed in the control pods, to a tank with a low pressure (for example of 100 psi or even less), we will not need to compensate for hydrostatic pressure of seawater as mandated by relating regulations and standards. But there is still the negative effect of riser pressure. The hydrostatic pressure of the riser exerts a force on the ram (depending on the cross sectional area of the shaft of the ram) and tends to push it back.

Some investigators are trying to move all BOP equipment to the surface to solve this problem.

In most cases, a properly sized rigid conduit to conduct power fluid from accumulators mounted on the surface to the BOP stack is capable of providing sufficient flow rates to operate BOP functions within API mandated limits, even in extreme water depths. However, certain BOP control systems must have dedicated sources of stored hydraulic fluid located on the BOP stack.

So, it would be better to stay with gas charged accumulators until a better alternative is developed.

NOMENCLATURE

A	Cross sectional area of piston in accumulators, $[L]^2$, in.^{2}
API	American Petroleum Institute
B	Magnetic flux density vector field, $[M][T]^{-2}[A]^{-1}$, gauss
BOP	Blowout Preventer
c	Spring Index
d	Wire Diameter, [L], in.
D	Mean diameter of cylindrical helical spring, [L], in.
d**F**	Differential force vector, $[M][L][T]^{-2}$, N
d**l**	Differential Length vector, [L], cm
D_p	Diameter of Piston in spring charged accumulator, [L], in.
D_w	Water depth, [L], ft
f	Deflection of spring, [L], in.
F	Magnetic Force vector, $[M][L][T]^{-2}$, N
F_S	Force of the spring, $[M][L][T]^{-2}$, lbf
G	Shear modulus of elasticity, $[M][L]^{-1}[T]^{-2}$, psi
g	Gravity Acceleration, $[L][T]^{-2}$, ft/s^2
h	Vertical displacement, [L], ft
I	Current, [A],Amps
k	Curvature correction factor
K&C	Kill and Choke
L	Length of shaft, [L], ft
m	Mass, [M], lbm
MMS	Minerals Management Service
NPD	Norwegian Petroleum Directorate
P	Load on the spring, $[M][L][T]^{-2}$, lbf
Pg	Pressure of gas, $[M][L]^{-1}[T]^{-2}$, psi
P_L	Pressure of hydraulic fluid, $[M][L]^{-1}[T]^{-2}$, psi
P_{max}	Maximum working pressure, $[M][L]^{-1}[T]^{-2}$, psi

P_{min} Minimum working pressure, $[M][L]^{-1}[T]^{-2}$, psi

P_n Precharge pressure, $[M][L]^{-1}[T]^{-2}$, psi

PTFE Polytetrafluoroethylene (Teflon)

R Rate of spring (Coefficient of spring), $[M][T]^{-2}$, lbf/in.

S Shearing Stress, $[M][L]^{-1}[T]^{-2}$, psi

S_v Safe shearing stress, $[M][L]^{-1}[T]^{-2}$, psi

S_v' Corrected shear stress, $[M][L]^{-1}[T]^{-2}$, psi

U Energy stored in spring, $[M][L]^2[T]^{-2}$, Joule

U_p Energy capacity per unite volume of spring material, $[M][L]^2[T]^{-2}$, Joule

V Volume of spring material, $[L]^3$, ft^3

V_{ac} Actual volume of accumulator, $[L]^3$, gal.

V_c Volume of gas at the maximum working pressure, $[L]^3$, gal.

V_d Volume of gas at the minimum working pressure, $[L]^3$, gal.

V_{left} Volume of liquid left inside a discharged accumulator, $[L]^3$, gal.

V_n Volume of gas at P_n, $[L]^3$, gal.

V_p Volume of fluid pumped into the accumulator to get P_{max}, $[L]^3$, gal.

V_U Usable Fluid, $[L]^3$, gal.

W Energy, $[M][L]^2[T]^{-2}$, Joule

w Weight, $[M][L][T]^{-2}$, lbf

x_0 Position of the larger piston at P_{max} in gas charged pressure intensifier, [L], in.

x_1 Position of the larger piston at P_{min} gas charged pressure intensifier, [L], in.

x_2 Position of the larger piston at P_n in gas charged pressure intensifier, [L], in.

Z Gas compressibility factor

Z_{max} Gas compressibility factor at P_{max}

Z_{min} Gas compressibility factor at P_{min}

Z_n Gas compressibility factor at P_n

θ The angle between d**l** and **B,** degree

REFERENCES

1. Good, A.C. and McAdams, P.J.: "Mathematical Prediction and Experimental Verification of Deep Water Accumulator Capacity" paper OTC 13234 presented at the 2001 Offshore Technology Conference, Houston, Texas, 30 April - 3 May.

2. *RP53, Recommended Practices for Blowout Preventers Equipment Systems for Drilling Wells*, 3rd edition, API, Washington, DC (1997).

3. Sattler, P.S.: "BOP Subsea Hydraulic Accumulator Energy Availability, How To Ensure You Have What You Need," paper SPE 74469 presented at the 2002 IADC/SPE Drilling Conference, Dallas, Texas, 26-28 February.

4. Curtiss, J.P., and Buckley M.: "Subsea Accumulators – Are they a False Reliance?," paper SPE 79881 presented at the 2003 IADC/SPE Drilling Conference, Amsterdam, The Netherlands, 19-21 February.

5. *Hydraulic Accumulator Products, Catalog HY10-1630/US*, Parker Hannifin Corp., Rockford, Illinois (2003).

6. *Hydac International Accumulator, Bladder, Piston, Diaphragm*, Hydac Corp. Accumulator Division, Bethlehem, Pennsylvania (2002) 5.

7. "NXT Double BOP", http://www.varco.com/products/c3shafer_3.htm, 10 June 2004.

8. "Pressure Control While Drilling System", http://www.varco.com/c3shafer_28.htm, 18 September 2004.

9. "Type 72 Shear Ram", http://www.varco.com/products/VDE/VDE.htm, 11 June 2004.

10. "DL annular Blowout Preventer", http://www.coopercameron.com, 13 June 2004.

11. Childers, M.A., Buckley M. and Curtiss, J.P.: "Fast Response Retrofitable Ultra Deepwater BOP Control System," paper SPE 87156 presented at the 2004 IADC/SPE Drilling Conference, Dallas, Texas, 2-4 March.

12. *Spec. 16D*, Specification for Control Systems for Drilling Well Control Equipment, API, Washington, DC (1993).

13. Sheffield, R., *Floating Drilling: Equipment and Its Use*, Gulf Publishing Co., Houston, Texas (1980).

14. Schlater, D.R. and Stanton, P.N.: "The Effect of Hydrostatic Pressure on The Operation of Subsea Blowout Preventers," paper SPE 8133 presented at the 1978 European Offshore Petroleum Conference & Exhibition, London, 24-27 October.

15. *Marks' Standard Handbook for Mechanical Engineers*, Baumeister, T., Avallone, E.A, Baumeister III, T., eighth edition, McGraw-Hill Book Co., Inc., New York City (1978).

16. *Spring Design Manual*, second edition, Society of Automotive Engineers, Inc., Warrendale, Pennsylvania (1996).

17. Wahl, A.H., *Mechanical Spring*, second edition, McGraw-Hill Book Co., Inc., New York City (1963).

18. Carlson, H., *Spring Designer's Handbook*, Marcel Dekker, Inc., New York City (1978).

19. Keenan, J.H., *Thermodynamics*, John Wiley & Sons, Inc., New York City (1963).

20. Ellis R.: "Exploring the Possibilities of a Naval Electromagnetic Rail-Gun", paper presented at 2003 Annual Gun & Ammunition Symposium, Austin, 24-27 March.

21. IDA, N. and Bastos, J.P.A., *Electromagnetics and Calculation of Fields*, second edition, Springer-Verlag Inc., New York City (1997).

22. "Design of an integrated electromagnetic levitation and guidance system for SwissMetro", http://leiwww.epfl.ch/publications/darrigo_rufer_epe_99.pdf, 13 September 2004.

23. Watson, D., Brittenham, T. and Moore, P.L.: *Advanced Well Control*, Monograph Series, SPE, Richardson, Texas (2003).

APPENDIX A

DERIVATION OF USABLE FLUID FORMULAS

Compressibility Factor Model. The procedure is the same as what we have done to derive Eq. 3.10. But for this model, we have to incorporate Z-factor into the gas equation of state. We have to calculate the volume of stored fluid and also, the volume of fluid released from the accumulator at the time it has been discharged. To get these values, we should calculate the volume of gas and subtract it from the actual fluid capacity of accumulator.

Eq. A.1 is known as the gas equation of state:

$$\frac{P_n V_n}{Z_n R T_n} = \frac{P_{\max} V_c}{Z_{\max} R T_{\max}}, \quad \text{(A.1)}$$

Since, changes in temperate are neglected, Eq. A.1 can be written as below:

$$\frac{P_n V_n}{Z_n} = \frac{P_{\max} V_c}{Z_{\max}}, \quad \text{(A.2)}$$

We solve the Eq. A.2 for V_c:

$$V_c = \left(\frac{P_n / Z_n}{P_{\max} / Z_{\max}} \right) V_n, \quad \text{(A.3)}$$

Recalling Eq. 3.3 and substituting V_{ac} for V_n, we can calculate the volume of fluid pumped into the bottle, V_p:

$$V_p = \left(1 - \frac{P_n / Z_n}{P_{\max} / Z_{\max}}\right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (A.4)$$

We follow the same procedure and calculate volume of fluid left inside the bottle after it has been discharged to the minimum pressure P_{min}:

$$V_{left} = \left(1 - \frac{P_n / Z_n}{P_{\min} / Z_{\min}}\right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (A.5)$$

It is obvious that the difference between V_{left} and V_p is volume of fluid recovered from accumulator or usable fluid. So, from Eq. A.5 and Eq. A.4 we have:

$$V_U = \left(\frac{P_n / Z_n}{P_{\min} / Z_{\min}} - \frac{P_n / Z_n}{P_{\max} / Z_{\max}}\right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (A.6)$$

where V_U is accumulator's usable fluid.

Adiabatic Expansion Model. In adiabatic process, the gas does not gain or lose heat energy. At higher pressures, the expansion of gas inside the accumulator is fast enough to consider the expansion of the gas as an adiabatic process. We use the following relation to calculate the volume of gas at minimum and maximum working pressure.

$$P_1 V_1^{\,n} = P_2 V_2^{\,n}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (A.7)$$

The constant n is determined by the characteristics of the precharge gas. The Power n is determined to be 1.4 for nitrogen.

In adiabatic method, the usable fluid is calculated by Eq. A.8:

$$V_U = \left(\left(\frac{P_n}{P_{\min}} \right)^{1/n} - \left(\frac{P_n}{P_{\max}} \right)^{1/n} \right) V_{ac}, \quad \text{(A.8)}$$

Let’s first calculate the volume of fluid we have to pump into the accumulator bottle to charge it to its maximum working pressure. The volume of gas at the maximum working pressure could be calculated by equation below:

$$P_n V_n^{\ n} = P_{\max} V_c^{\ n} \Rightarrow V_c = \left(\frac{P_n}{P_{\max}} \right)^{1/n} V_n, \quad \text{(A.9)}$$

We subtract V_c from actual fluid capacity of accumulator, V_{ac}, to get the volume of fluid stored in the accumulator, V_p. Recalling $V_{ac}=V_n$, we will have:

$$V_p = \left(1 - \left(\frac{P_n}{P_{\max}} \right)^{1/n} \right) V_{ac}, \quad \text{(A.10)}$$

Now, the volume of gas at the minimum working pressure can be calculated by equation below:

$$P_n V_n^{\ n} = P_{\min} V_d^{\ n} \Rightarrow V_d = \left(\frac{P_n}{P_{\min}} \right)^{1/n} V_n, \quad \text{(A.11)}$$

Knowing the volume of gas at the minimum working pressure and actual fluid capacity of the accumulator, V_{ac}, we can calculate the volume of fluid left inside the accumulator after it has been discharged. We substitute V_{ac} for V_n in Eq. A.11 and subtract V_d from V_{ac} to get the volume of fluid left inside the bottle, V_{left}:

$$V_{left} = \left(1 - \left(\frac{P_n}{P_{\min}}\right)^{1/n}\right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(A.12)}$$

Obviously the difference between V_{left} and V_p should be the usable fluid.

$$V_U = \left(\left(\frac{P_n}{P_{\min}}\right)^{1/n} - \left(\frac{P_n}{P_{\max}}\right)^{1/n}\right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(A.13)}$$

Since n is 1.14 for nitrogen, we have:

$$V_U = \left(\left(\frac{P_n}{P_{\min}}\right)^{0.71} - \left(\frac{P_n}{P_{\max}}\right)^{0.71}\right) V_{ac}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(A.14)}$$

APPENDIX B

SPRING DESIGN FORMULA DERIVATION

The theoretical treatment given involves three different methods to derive design formulas for round wire helical compression spring: 1) elementary theory, curvature and direct shear effects being neglected, 2) approximate theory, considering such effects, and 3) exact analysis, based on the theory of elasticity.[17]

In this appendix, we try to derive design formula based on elementary method. This theory is based on the assumption that an element of an axially loaded helical spring behaves essentially as a straight bar in pure torsion (**Fig. B.1**).[17]

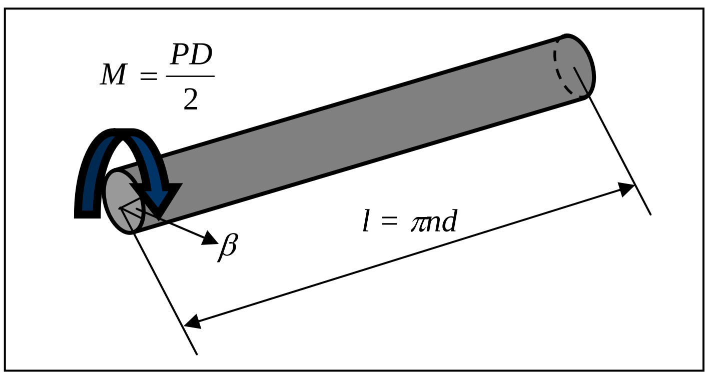


Fig. B.1- Straight bar as a helical spring.

Referring to Fig. B.1, each element of the spring coil is assumed to be subject to torque $PD/2$ acting about the center, where D is the mean diameter of the spring, and P is the load on it (**Fig. B.2**). Assuming that planes perpendicular to the bar axis do not rap or distort during deformation, it follows that the shearing deformation, and hence the shearing stress, will have a linear distribution along a radius as shown in **Fig. B.3**.[17] Hence, at a distance ρ from the center O the shearing stress will be $\tau_\rho = 2\rho\tau/d$ where τ is maximum shearing stress at the surface of the bar and d = bar diameter (Fig. B.3). The moment, dM, taken up by a ring of width $d\rho$ at a radius ρ can be determined as follows:

$$dF = \tau_{\rho}\, dA, \quad \text{(B.1)}$$

Since the area of the ring, *dA*, is $2\pi\rho d\rho$, we have:

$$dF = 2\pi\tau_{\rho}\rho\, d\rho, \quad \text{(B.2)}$$

Base on definition of moment (arm × force), we have $dM = dF \times \rho$. So,

$$dM = \frac{4\pi\rho^3\tau}{d} d\rho, \quad \text{(B.3)}$$

Integrating both sides of Eq. B.3, we would get:

$$M = \int_0^{d/2} \frac{4\pi\tau\rho^3}{d} d\rho = \frac{\pi d^3 \tau}{16}, \quad \text{(B.4)}$$

Since the total moment *M* is equal to *PD/2*, we replace it in Eq. B.4 by *PD/2* and solve it for τ:

$$\tau = \frac{8PD}{\pi d^3}, \quad \text{(B.5)}$$

This is the uncorrected stress (Eq. 4.4), which neglects effects due to bar curvature and direct shear stress.[17]

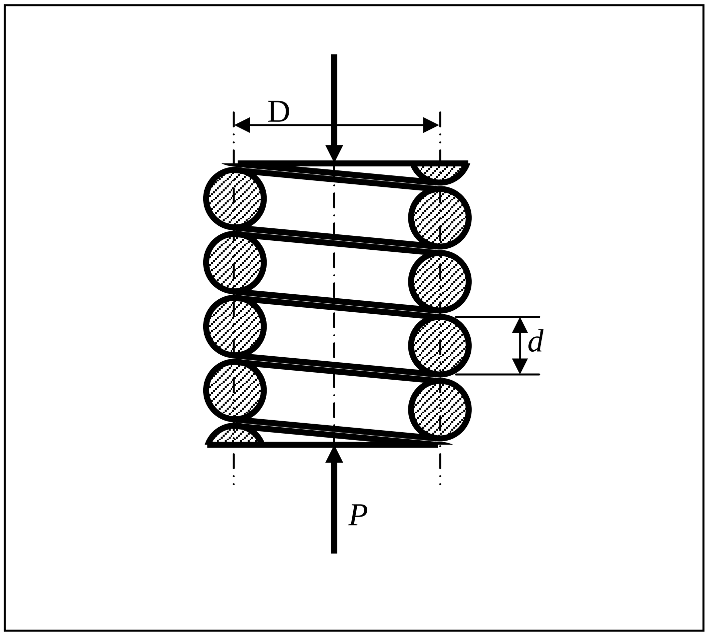


Fig. B.2- Helical compression spring axially loaded.

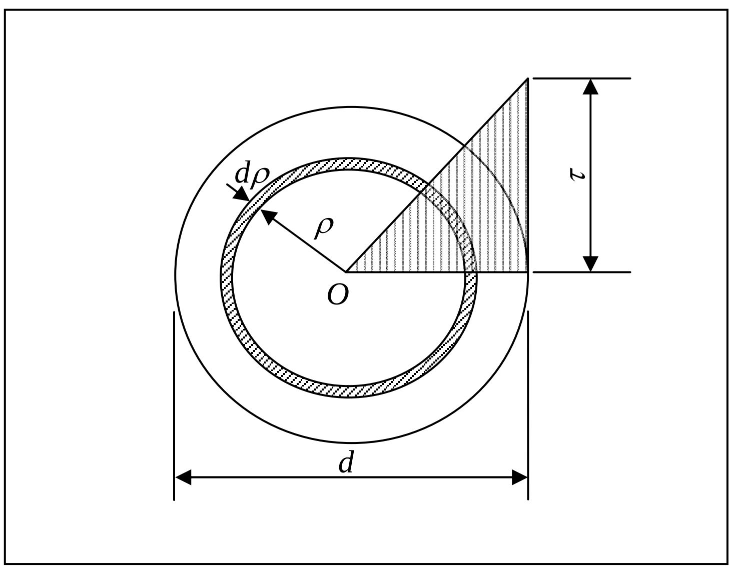


Fig. B.3- Cross sectional element of spring under torsion.

To calculate deflection of a helical round wire compression spring, the following procedure may be employed. Consider an element *ab* on the surface of the bar and parallel to the bar axis (**Fig. B.4**)

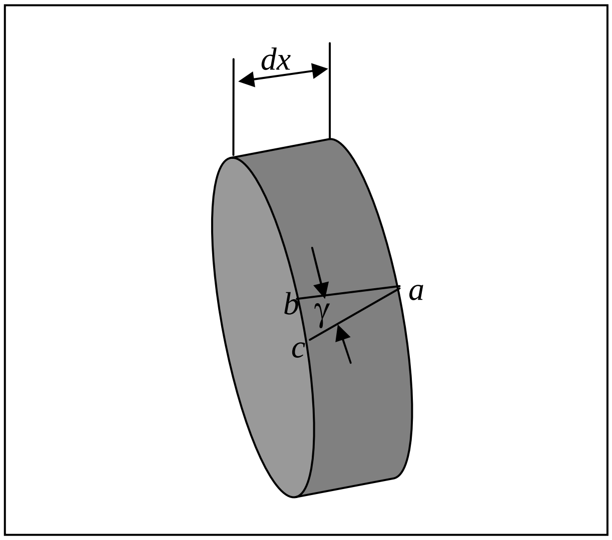


Fig. B.4- Element *ab* on the surface of the bar.

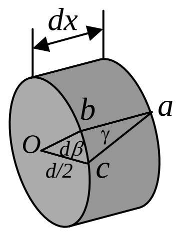
After deformation, this element will rotate through a small angle γ to the position *ac*. From elastic theory, this γ will be equal to:[17]

$$\gamma = \frac{\tau}{G}, \quad \text{(B.6)}$$

Thus, from Eq. B.5:

$$\gamma = \frac{8PD}{\pi\, d^3 G}, \quad \text{(B.7)}$$

Since the distance $bc = \gamma\, dx$ for small angles such as we are considering, the elementary angle $d\beta$ through which one cross section rotates with respect to the other will be equal to $2\gamma\, dx/d$ (**Fig. B.5**).



$bc = \gamma\, dx$, and
$bc = (d/2)\, d\beta$, $\Rightarrow$ $\gamma\, dx = (d/2)\, d\beta$

$\Rightarrow$ $d\beta = 2\gamma\, dx / d$

Fig. B.5- Elementary angle $d\beta$.

Again assuming that the spring may be considered as a straight bar (Fig. B.1) of length $l = \pi n D$ where n is number of active coils. The total angle β, representing the angular deflection of one end of the bar with respect to the other, will be:

$$\beta = \int_0^{\pi n D} \frac{2\gamma dx}{d} = \int_0^{n\pi D} \frac{16PDdx}{\pi d^4 G} = \frac{16PD^2 n}{Gd^4}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(B.8)}$$

Since the effective moment arm of the load is equal to D/2, the deflection at the load will be:[15]

$$f = \beta \frac{D}{2}, \quad \text{(B.9)}$$

Now, we replace β in Eq. B.9 by Eq. B.8 to get total deflection in helical compression spring:

$$f = \frac{8PD^3 n}{Gd^4}, \quad \text{(B.10)}$$

This is the commonly used formula for spring deflections. In contrast to the stress formula, which may be in considerable error due to curvature effect, this is quite accurate (within 1 or 2%) even for fairly small spring indices.[17]

The elementary method discussed above yields an average stress around the periphery of the spring cross section. However, the maximum stress occurs at the inside of the coil (**Fig. B.6**) may be considerably higher than that given by Eq. B.5 by an amount depending on the spring index *D/d*.[17] The reason for this is the short fiber length *a'b'* at the inside of the coil (**Fig. B.7**). Thus, if the redial sections *bb'* and *aa'* rotate through a small angle with respect to each other and about the bar or wire axis, the inside (and the shorter) fiber *a'b'* will be subject to a much higher shearing strain than the outside fiber *ab*. Also at the inside of the coil, the shear stress due to the direct axial load *P* is added to that due to the torque moment *PD/2* at this point ($4.29P/\pi d^2$).[17] Approximate theory considers these issues and derives the shearing stress formula at *a'* which is the maximum shearing stress occurred in spring bar due to compressing load *P* is:[17]

$$\tau' = \frac{8PD}{\pi d^3}\left(\frac{4c-1}{4c-4} + \frac{0.615}{c}\right), \quad \text{(B.11)}$$

where *c* is *D/d*.

This is the commonly used formula for corrected stress. In design of the spring, we have to make sure that this corrected stress does not exceed from the safe shear stress of the bar when it is completely deflected to its solid length at the maximum load *P*.

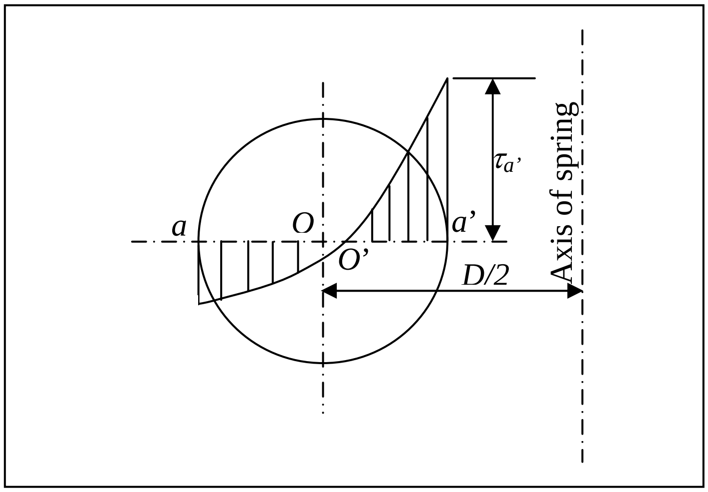


Fig. B.6- Stress distribution across the spring wire.

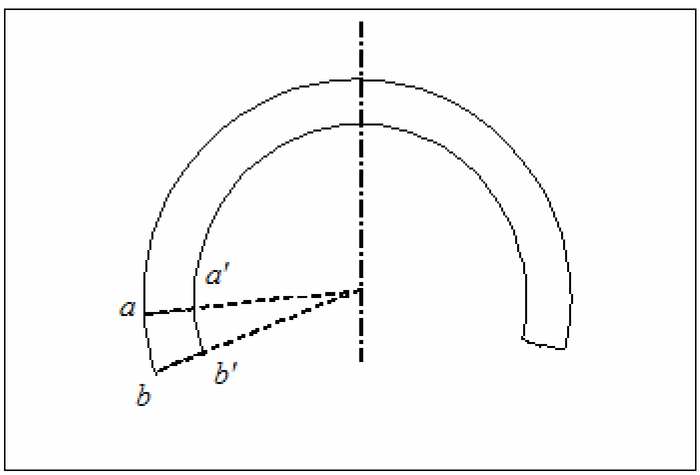


Fig. B.7- Shorter fibers inside the coil.

APPENDIX C

SPRING MATERIALS

High-Carbon spring steel

These materials are for springs that are cold coiled, with the material at room temperature. These include music wire, oil-tempered wire, hard-drawn carbon, valve spring-quality steels, and high carbon valve spring steel.

Music wire. The name is derived from the application of this type of wire to stringed instruments, especially the piano. This is a high quality carbon steel, but this wire is widely used for small-sized helical springs, particularly those subject to severe stress conditions.[17] The high strength of this material is obtained by using a steel of about 0.70 to 1% carbon, cold drawing to size using small steps with special annealing at certain points in the process.

The elastic limit, as a percentage of the ultimate strength, for music wire in torsion is 45 to 50%. The tensile strength of these wires varies considerably with respect to bar diameter; 440,000 psi for wires thinner than 0.020 in. and 230,000 psi for 0.220 in.[18] For design purposes, the modulus of elasticity G in torsion as used for compression spring is 11,200,000 psi for wire diameters up to 0.032 in. bars. For other sizes of its bar you can refer to **Table C.1**.

Table C.1- Shearing modulus of elasticity of music wire.[18]

Wire Diameter, in.	Shearing Modulus of Elasticity, psi
Up to 0.032	11,2000,000
0.033 – 0.063	11,850,000
0.064 – 0.125	11,750,000
Over 0.125	11,600,000

Hard-drawn spring steel. Hard-drawn spring steels, while of lower quality than music wire or oil-tempered wire, nevertheless is an important spring steel where cost is an important factor and where long fatigue life and uniformity are of lesser important. This wire may contain surface defects such as hairline seams.[17] It is best suited for those applications where long life and accuracy of loads are not too important.[18] Diameters available run up to small size of 0.163 in.

Recommended design stress for compression springs is 85,000 psi for wire diameters of 0.500 in. and this limit increases as wire diameter decreases; 145,000 psi for wire diameter of 0.020 in.[18]

Table C.2 gives you G evaluated for different diameters of these wires.

Table C.2- Shearing modulus of elasticity of hard drawn spring steel.[18]

Wire Diameter, in.	Shearing Modulus of Elasticity, psi
Up to 0.032	11,7000,000
0.033 – 0.063	11,600,000
0.064 – 0.125	11,500,000
Over 0.125	11,400,000

Oil-tempered spring steel. Oil-tempered spring steels are good quality, high carbon spring steels, uniform in quality and temper, and used for a large majority of coil springs requiring wire diameters from 0.125 to 0.5 in. These materials are used only for cold-coiled springs.[18]

Elastic limit of oil-tempered spring steel as percentage of its tensile strength is 45 to 50% for wire diameters of up to 0.25 in. and for wire diameters of 0.25 in. and larger is 40 to 45%. Tensile strength of this material changes as wire diameter changes; 190,000 psi for 0.500 in. wires to 300,000 psi for wire diameter of 0.200 in.[18]

Shearing modulus of elasticity of these wires has been evaluated to be 11,200,000 psi.[18]

Valve spring-quality steel. This alloy-steel wire has been frequently specified for cases such as aircraft engine valve springs where a high quality material is needed and where temperatures are somewhat higher than normal.[18] Since this material has been the most popular material in common use for automotive engine valve springs, it is called valve spring steel.

High carbon valve spring steel. This steel is the highest quality of round carbon steel valve spring wire available. It is uniform in quality and temper, but available in a limited range of diameters from 0.062 to 0.250 in. only, although other sizes can be obtained if the quantity desired is sufficient to be of interest to wire mills.

Alloy spring steel

The alloy spring steels have an important place in the field of spring materials, particularly for conditions involving high stress and where shock or impact loadings occur. All these alloys are used in springs for machines and aircraft equipment in wire sizes frequently under ¼ in. diameter and up to 0.500 in., but annealed bars are available in larger diameters from 3/8 In. to 2 in. or larger. These larger sizes, in the annealed condition, are hot rolled into heavy springs for armament and heavy equipment. These materials are for both springs that are cold-coiled, with the material at room temperature, and for hot-coiled springs. Alloy spring steels include chromium-vanadium steel spring wire, Chromium-Vanadium valve spring steel, and Chromium-Silicon alloy spring steel wire.

Chromium-vanadium spring steel wire. chromium-vanadium spring steel is a good quality, fairly high carbon, alloy spring steel containing small amounts of chromium and vanadium to increase the hardness, tensile strength, and endurance properties. It is obtainable in the round or square sections from 0.020 to 0.500 in. diameters for general use and in heavier sizes up to 2 in. and larger for hot-rolled springs. This wire is a popular alloy spring steel for applications involving higher stresses than can be used with the high carbon spring steels and for springs subjected to impact or shock, such as in pneumatic hammers. This material is also for moderately elevated temperature application up to 425 °F.[18]

Shearing modulus of elasticity is considered to be 11,200,000 psi in compression spring design. Recommended design stresses, depending on the condition of application and wire sizes, varies from 165,000 psi in average services for wire diameter of 0.020 in. to 120,000 psi for wire diameter of 0.500 in.[18]

Chromium-vanadium valve spring steel. It is a high quality alloy steel used for valve springs and can be obtained in variety of sizes from 0.02 to 0.50 in. This material is used only for cold-coiled springs.[18]

Elastic limit for this material is considered to be 100,000 to 130,000 psi in compression spring design.[17] Shearing modulus of elasticity has been evaluated to be 11,200,000 psi.

Chromium-silicon alloy spring steel wire. This alloy is a special alloy spring steel wire especially suited for highly stressed springs subjected to shock or impact loading such as recoil springs in antiaircraft guns. It is also used for moderately high temperature conditions up to 475 °F.[18] Diameters from 0.032 to 0.437 in. are available.[17]

Elastic limit for bars made of this material is in a range of 130,000 to 160,000 psi.[17] Recommended stress for design purposes decreases as bar diameter decreases; for 0.500 in. diameter bars, this stress may be evaluated to be 125,000 psi in average services and for 0.020 in. diameter bars is to be 165,000 psi.[18] G can be 11,200,000 psi in compression spring design calculations.

Hot-Rolled bars for hot-coiled springs

The large majority of springs in industry, especially those having wire sizes less than 3/8 in., are cold-coiled with the material at room temperature. If the spring index, *D/d*, is above 6, cold-coiling can be accomplished with spring having wire diameters up to 5/8 in.[18] For smaller indexes, and heavier wire or bars, the customary practice is to heat the bars to a distinct red heat, often above the hardening temperature, and hot-coil the spring. Steel is quite easy to coil or form while red hot.

For design purposes, the modulus of elasticity G in bending as used for compression spring is 10,750,000 psi.

Hot-rolled carbon steel bars. These bars are available up to 2 in. or larger.[18] This plain high carbon spring steel is the most widely used of all materials for hot-coiled springs because of its low cost and general availability. It has been used for springs in railway cars, trucks, buses, automobiles, ships, and as a buffer and safety springs for elevators. In heavy sections this steel has rather poor hardenability and lowered mechanical properties, particularly to resistance against shock loading, overloads, and settling in service.

The elastic limit, as a percentage of the ultimate strength, for high carbon steel bars in torsion is 50 to 60%. The tensile strength of these bars varies from 175,000 to 195,000 psi.

Hot-rolled alloy steel bars. These steel bars are especially useful for springs subjected to shock or suddenly applied loads and also are used continuously at elevated temperatures up to 800 °F and up to 950 °F intermediately.[18] These steel bars available in various types such as chromium alloy, silicon-manganese alloy, chromium-molybdenum alloy steel, and nickel-chromium-molybdenum alloy steel bars. These alloys are obtained in large diameters; nickel-chromium-molybdenum is available in 2.5 in. SAE 8660 of this family is especially recommended for all sizes over 1 in.[18]

The elastic limit for these alloy steel bars as used for design purposes in compression springs, as a percentage of the ultimate strength, is 60 to 70%. The tensile strength of these bars varies from 180,000 to 200,000 psi.

Non-ferrous spring material

These materials are used in cases where electrical conductivity or corrosion resistance is desired. For example Phosphor-bronze of this family is used for its good electrical conductivity and corrosion resistance, and Monel, another member of this family, is used in food industry. Phosphor-bronze material along with Spring Brass, Nickel-silver, Silicon-bronze, and Beryllium-copper have high percent of copper in their composition; more than 55% in Spring Brass to 97% in Silicon-bronze.

Monel, “K” Monel, Permanickel, Inconel, Inconel “X”, and Ni-span-C have nickel as the major element of their composition (42 to 97%). These all materials are cold-rolled.

APPENDIX D

GAS CHARGED ACCUMULATOR ENERGY

The work done by a perfect gas during a reversible adiabatic expansion can be calculated by the following formula:[19]

$$W = \frac{P_1 V_1}{n-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\left(1-\frac{1}{n}\right)}\right], \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. \quad \text{(D.1)}$$

We can use this formula for accumulators by approximate. Obviously the energy recovered from compressed gas in the structure of an accumulator is not exactly the same as what we get from Eq. D.1 due to energy dissipation during its discharge to the minimum working pressure; gas expansion in accumulator is not a 100% reversible adiabatic process. However this equation is accurate enough for our analysis.

The work done by the accumulator mentioned in chapter III, as example, at the depth of 7,000 ft is approximately equal to 3.17×10^6 Joules, where $P_1 = P_{max}$, $V_1 = V_c$, $P_2 = P_{min}$, and n is equal to 1.41.

Usable Fluid of that accumulator was also calculated at the depth of 7,000 ft to be 2.2 gallons. As you can see in Table 2.1, we need 51.1 gallons of fluid just to close one unite of annular preventer. So, we need 24 accumulators to be discharged to their minimum working pressure to do this function (51.1 / 2.2 = 23.23). Since every one of those accumulators can store 3.17×10^5 Joules, total energy recovered from 24 accumulators would be 7.36×10^6 Joules ($24 \times 3.17 \times 10^5$). So we need 7.36×10^6 Joules energy to do that function on the BOP stack through hydraulic BOP control system.

APPENDIX E

GAS CHARGED PRESSURE INTENSIFIER

In intensifiers a higher pressure can be generated by a lower pressure. We are trying to analyze the possibility of an accumulator which is technically similar to intensifiers. In a gas charged pressure intensifier, a higher pressure, for example 8,000 psi, can be generated by a gas pressure of much lower than 8,000 psi, for example 3,000 psi at which nitrogen behaves like an ideal gas.

A gas charged pressure intensifier may look like **Fig. E-1**. Fig. E-1 shows that a larger piston is required to be connected to a smaller piston by a shaft to generate a higher pressure. If we neglect the forces on pistons which are exerted by the very low pressure gas inside the low pressure room of this kind of accumulator, we can write;

$$P_g A_g = P_L A_L, \quad \text{………………………………………..………………………} \quad \text{(E.1)}$$

To get a pressure of 8,000 psi in the liquid side of this accumulator from pressure of 3,000 psi (the pressure of gas in the gas side of the accumulator) the ratio of piston areas, A_g/A_L, should be 8/3, which is the ratio of the pressures mentioned above. For example; if the area of smaller piston was 50.27 in^2, the area of the larger piston should be 134.04 in^2, which is really a large piston.

What about the height of such an accumulator, if we are interested in a usable fluid volume of 15 gal.? Considering the area of smaller piston (50.27 in^2), the stroke of the piston between the maximum working pressure and minimum working pressure in liquid side of our tentative accumulator should be 68.93 in. to release a usable fluid volume of 15 gal. (15 gal.×231/50.27 in^2 = 68.93 in.). So, obviously the stroke of the larger piston would be 68.93 in.;

$$x_1 - x_0 = 68.93\,in., \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(E.2)}$$

From the gas equation of state we have;

$$P_{\max} x_0 = P_{\min} x_1, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(E.3)}$$

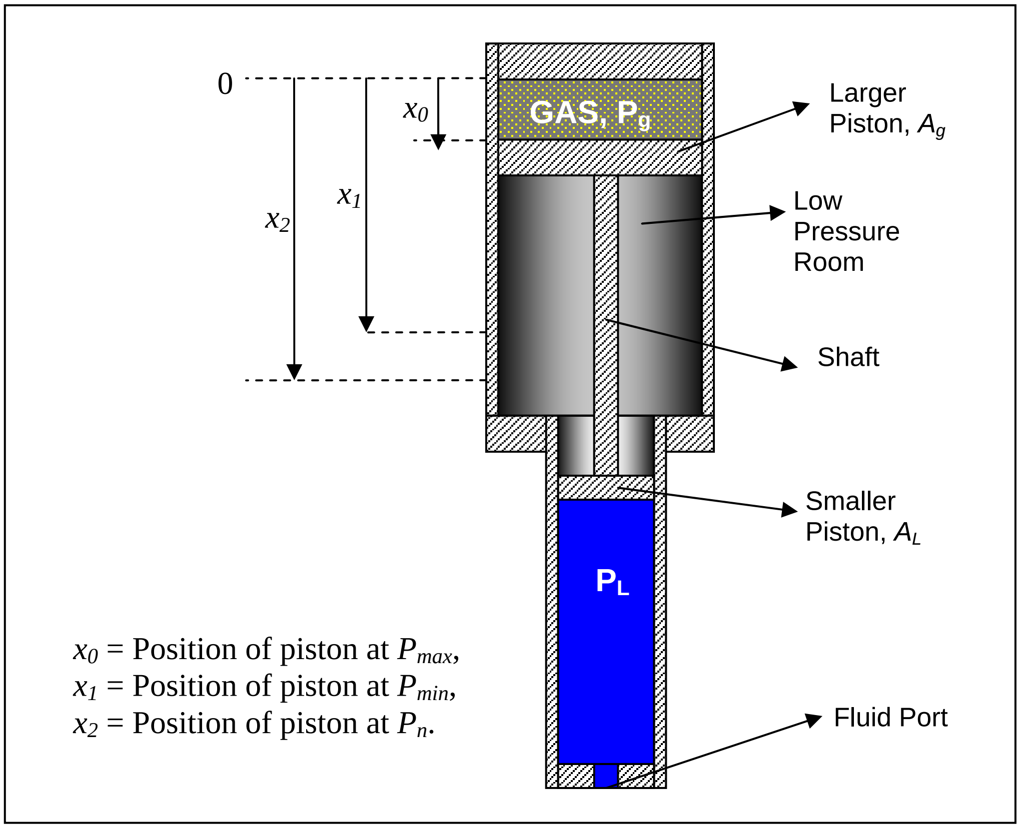


Fig. E.1- A gas charged pressure intensifier.

From Eq. E.2 and Eq. E.3 and for P_{min} = 1,200 psi, x_0 is calculated to be 45.95 in. Again from gas equation of state we have;

$$P_{\max} x_0 = P_n x_2, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad \text{(E.4)}$$

Using Eq. E.4, and P_n = 1,000 psi, x_2 is calculated to be 137.86 in. It means that the length of upper cylinder should be larger than 137.86 in. Obviously, the length of the lower cylinder also should be 137.86 in. So, the length of such an accumulator should be more than 275.72 in. which is taller than a 2-storey building. It is necessary to mention that we did not get into account the thickness of the gas cap and liquid gap, and a gas valve at the top of this accumulator. Clearly, the height of such an accumulator for a usable fluid of 15 gallon would be more than 275.72 in.

Here is the general formula for calculation of x_2;

$$x_2 = \frac{P_{\max}}{P_n} \times \frac{1}{\left(\frac{P_{\max}}{P_{\min}} - 1\right)} \times \frac{V_U}{A_L}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(E.5)}$$

For P_{max} = 3,000 psi, P_{min} = 1,200 psi, and P_n = 1,000 psi, we can simply write;

$$x_2 = 2\frac{V_U}{A_L}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(E.6)}$$

Where the unit of V_U is in^3 and A_L is in^2.

The volumetric efficiency of such an accumulator, on the other hand, like a conventional deep water accumulator is very low;

$$\text{Volumetric Efficiency} = \frac{\text{Usable Fluid}}{\text{Volume of Gas at } P_n}, \quad \ldots\ldots\ldots\ldots \text{(E.7)}$$

then;

$$\text{Volumetric Efficiency} = \frac{15\,\text{gal.} \times 231\,\text{in}^3 / gal.}{137.86\,\text{in.} \times 134.04\,\text{in}^2} \times 100 = \%18.75.$$

Volumetric efficiency of such an accumulator can be calculated simply by the equation below;

$$\text{Volumetric Efficiency} = \frac{1}{2}\frac{A_L}{A_g} \times 100, \quad \text{(E.8)}$$

However, it is concluded that this kind of accumulator is not practical, because of its length and low volumetric efficiency.

APPENDIX F

REGULATIONS AND STANDARDS

To size an accumulator bank, the volume of usable fluid required to manipulate the BOP stack components must be determined. The volume of usable fluid also should meet a regulatory agency requirement. So, the minimum required usable fluid volume depends on the BOP stack equipment operating volume and what the government regulations dictate.

The major standards and regulations that address accumulator volumes are the following;[3]

- API Recommended Practice 53,
- API Specification 16D,
- The Mineral Management Service (MMS),
- The Norwegian Petroleum Directorate.

Each of these standards has different requirements for hydraulic fluid volume.

API Recommended Practice 53. BOP system should have sufficient usable hydraulic fluid volume (with pumps inoperative) to close and open one annular-type preventer and all ram-type preventers from a full open position against atmospheric wellbore pressure. After closing and opening one annular preventer and all ram-type preventers, the remaining pressure shall be 200 psi or more above the minimum recommended precharge pressure.[2]

API Specification 16D. The BOP control system shall have a minimum stored hydraulic fluid volume, with pumps inoperative, to satisfy the greater of the two following requirements:

1. Close from a full open position at wellbore pressure, all of the BOP's in the BOP stack, plus 50% reserves.
2. The pressure of the remaining stored accumulator volume after closing all of the BOP's shall exceed the minimum calculated (using the BOP closing ratio) operating pressure required to close any ram BOP (excluding the shear rams) at the maximum rated wellbore pressure of the stack.[12]

Mineral Management Service. The MMS requirement in the United States federal waters is that a bank of accumulator be sized to provide 1½ times the fluid volume necessary to close all BOP equipment units with a minimum pressure equal to 200 psi above precharge pressure.[23]

Norwegian Petroleum Directorate. The NPD requirement for North Sea operators is more stringent. It requires an accumulator volume adequate to close, open, and close all preventers and thereafter leave a minimum pressure reserve equal to %25 of the amount of pressure it takes to close all BOPs. This must be done with all pumps inoperative and with a minimum pressure 200 psi over precharge pressure.[23]

Fig. F.1 shows the differences in the usable fluid requirements among the four major standards and regulations. The stack used for the calculations is comprised of one Cameron 10K DL annular and four Cameron 15K TL rams.

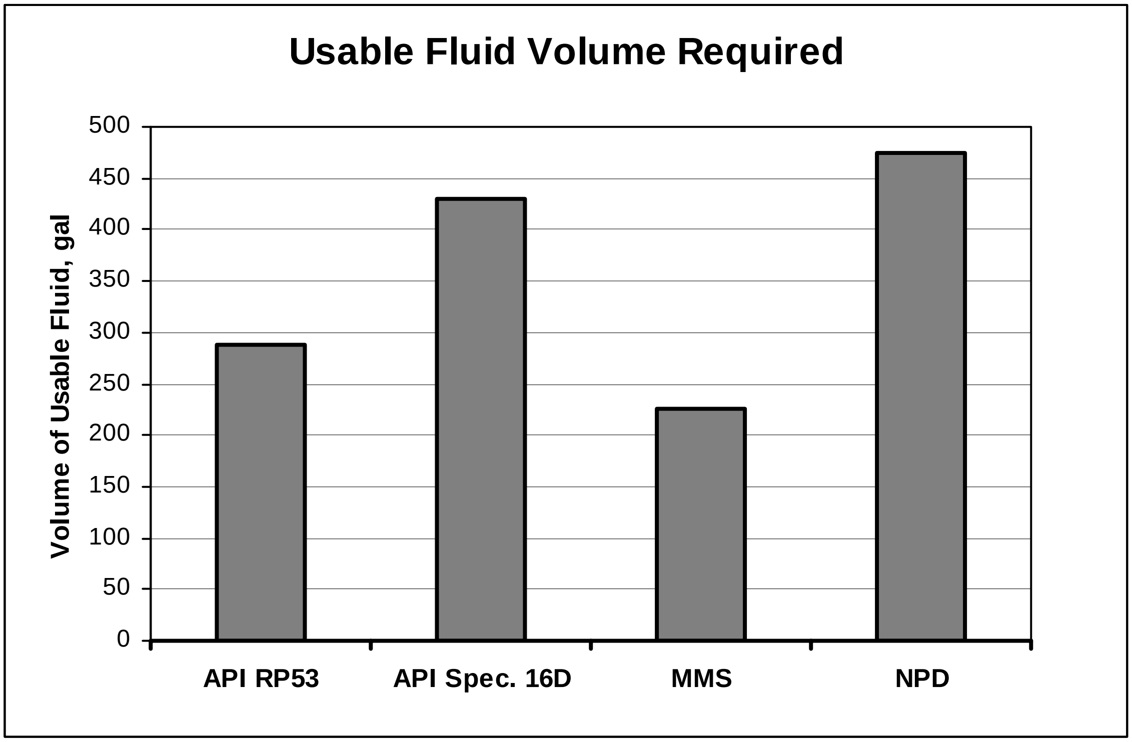


Fig. F.1- Comparison of different standards and regulations.[3]

VITA

Name:	Mehdi Mir Rajabi
Permanent Address:	#27, Mahmoudi St., Varamin, 33718 IRAN
Education:	B.Sc., Mining Engineering Sahand University of Technology Tabriz, IRAN (Jan. 1997 – Sep. 2001)
	M.Sc., Petroleum Engineering Texas A&M University College Station, TX, 77843 USA. (Jan. 2003 – Dec. 2004)